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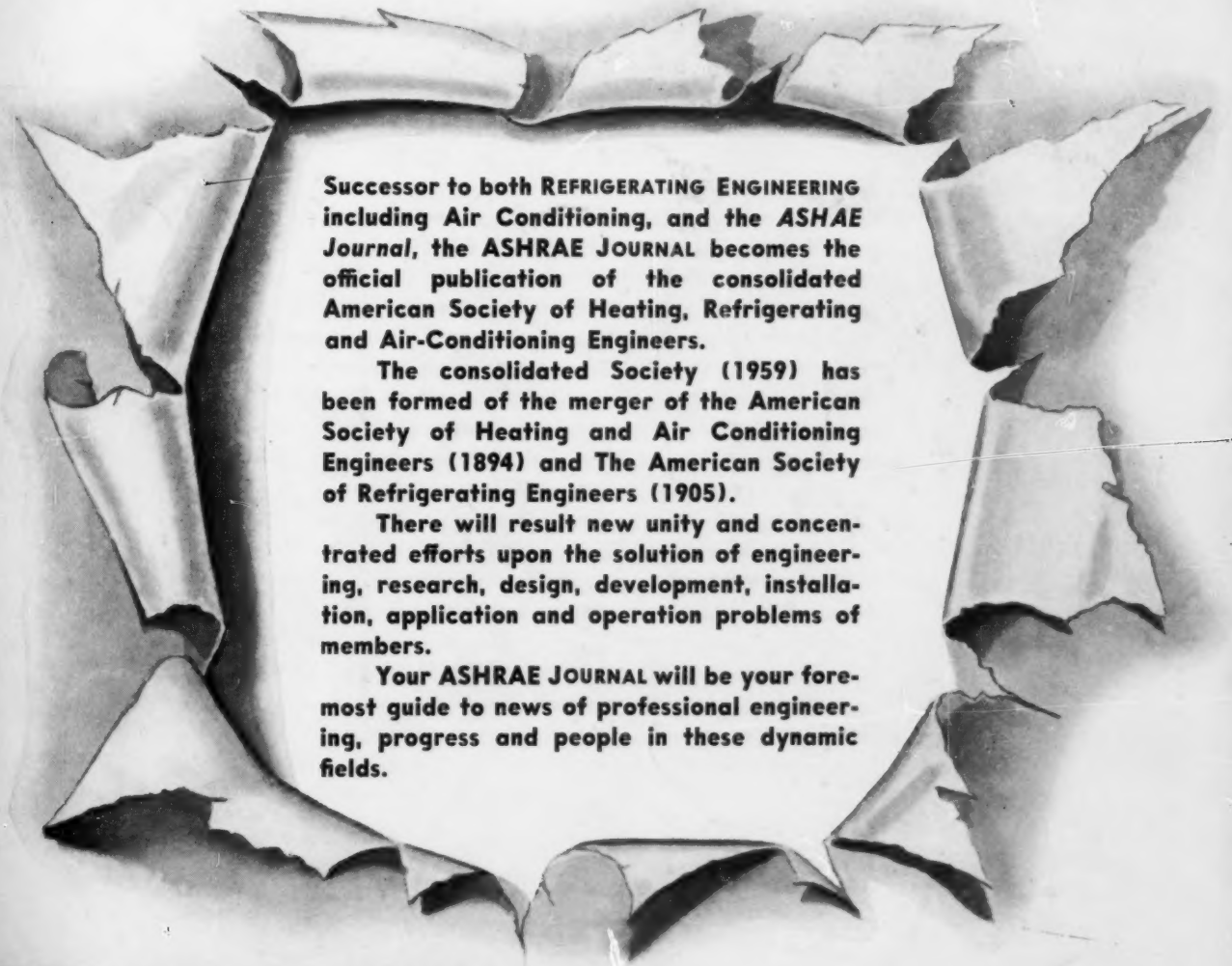
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS



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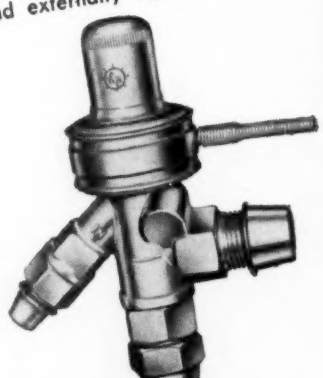

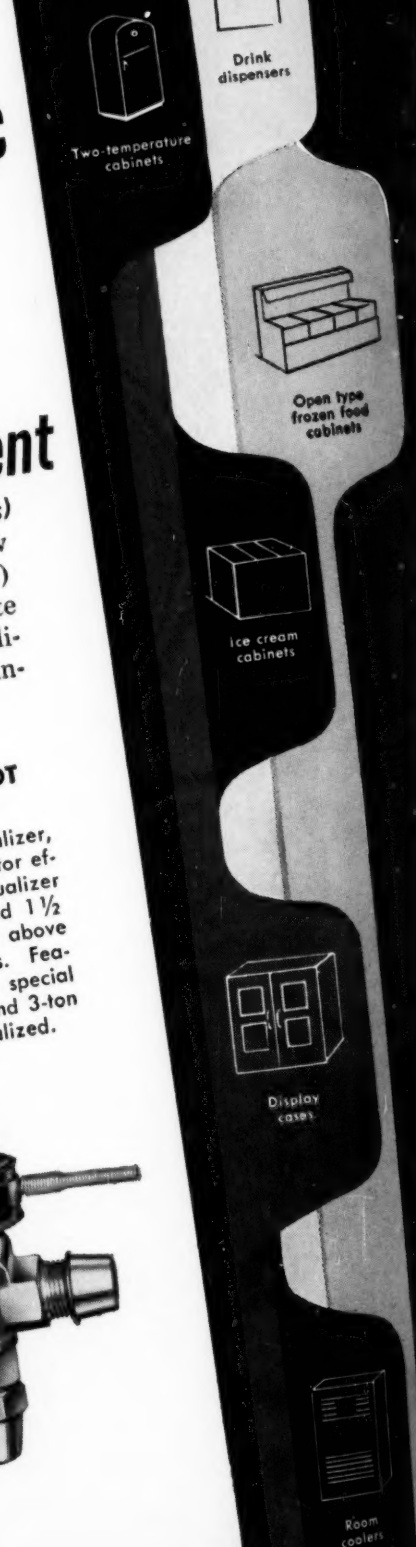

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Formerly Refrigerating Engineering including Air Conditioning, and incorporating the ASHAE Journal.

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Comment

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OFF TO A FAST START

"It is far better", someone has said "to light one little candle than to sit and curse the dark". Just such constructive objectivity has quite apparently led to a heart-warming unity of purposes and plans for the new American Society of Heating, Refrigerating and Air-Conditioning Engineers. Dissenters have joined with merger proponents in impressive array. We confidently predict a bright future for this newest of American engineering societies—one in which all groups and factions will join in further perpetuating efforts toward the better accomplishment of long-established goals.

With the conclusion of the 65th Annual Meeting of ASHAE in Philadelphia late in January, the last of the separate national meetings of either merged society has been held. The previously planned 55th Annual Meeting of ASRE at Lake Placid becomes the first of the annual meetings of ASHRAE, June 22-24. The first of the semiannual meetings of ASHRAE will be in Dallas, Texas, February 1-4, 1960.

Thus rapidly do the continuities establish themselves. Committees are being appointed and other steps anticipated in the Plan for Merger are being accorded effectiveness. Indeed, that which once seemed like a three or even four year project of unification may well be done in half the time. Such are the outcomes of willing hands and eager minds.

ABOUT THIS ENGINEERING—

In all the pros and cons about the merger which led to ASHRAE, there was one which impressed us with its irrefutable logic. Whether their activities had to do with heating or cooling, the members of both antecedent societies had at least two basic engineering phenomena with which to work—the transfer of heat and the circulation or maintenance of the transfer medium. That was the real common ground within engineering aspects. All else was so much icing on the cake.

From that base derive certain identities in the points of applications—living, working, merchandising, transportation and recreational zones. And the objectives of comfort, health, efficiency, processing and preservation.

Heavy emphasis, indeed, was placed upon the Research Program of ASHRAE in the formulation of staff and budgetary considerations allied therewith. The broad extent of this has been questioned but, as we see it, whether performed in the ASHRAE Laboratory or in independent places, the continuous augmentation of knowledge about the fundamental engineering that lies behind the whole concept of ASHRAE is an inescapably important one.

Edward R Searles
Editor

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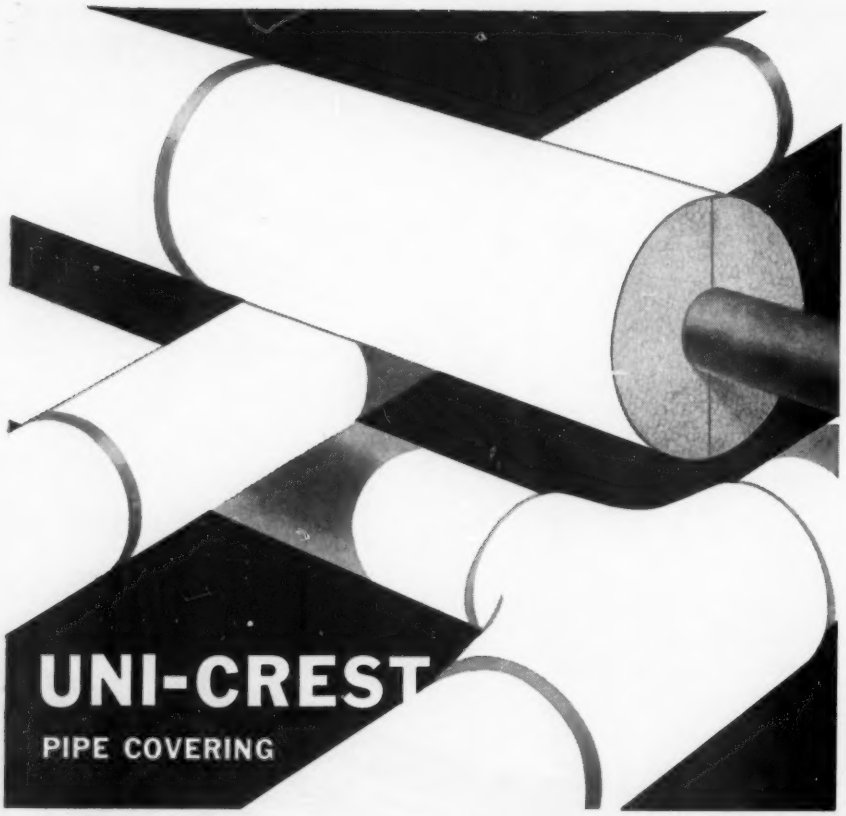

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I CAN NOT AGREE . . .

To the Editor:

In your article, REFRIGERATING ENGINEERING, January 1959 by A. S. Decker and S. Gianni, "Predicting Leaving Dry Bulb Temperatures for Air Conditioning Coils," I find a few points that should be made clear.

The authors state that the apparatus dew point "bears no resemblance to the actual surface temperature." In order to justify the statement they compare an ADP obtained from their experiment to the calculated surface temperature on the *leaving side* of the coil finding approximately 10F difference when the refrigerant temperature is 35F. It seems to me the statement is not prudent.

First, *the actual coil surface temperature* varies from row to row and from tube surface to fin surface. We have roughly measured in our laboratory the fin surface temperature of the coil by attaching thermocouples at both front and rear sides of the plate fin.

The approximate average fin surface temperature obtained by our thermocouple readings closely resembles the apparatus dew point determined by the air conditions before and after the coil. Moreover, the apparatus dew point is quite close to the computed average surface temperature determined by the writer's paper (Authors' Reference 5).

I can agree with the authors that the ADP is rather a fictitious entity by definition. But, I cannot agree with the authors' statement that the ADP bears no resemblance to the true surface temperature.

The writer is well aware that the surface temperature at the fin root, particularly at the leaving air side of the coil, is very close to the refrigerant temperature. (But it is not fair to compare the local surface temperature such as that at the leaving air side of the coil to ADP, because ADP closely resembles the average surface temperature.)

Perhaps the most interesting point of the paper is that the authors have found a cooling curve in their experiment which is independent of the coil geometry and the air face velocity. But this is fairly obvious to me without involving the kinetic theory of gases (as the authors suggested). The basic mass and heat transfer differential equation over the dehumidifying surface does not contain the coil geometry and face velocity. Thus, the solution of the differential equation should yield a unique cooling curve as long as dimensionless factors involved in the solution are maintained similar (which may have been the case of the authors' experiments).

The important problem involved in

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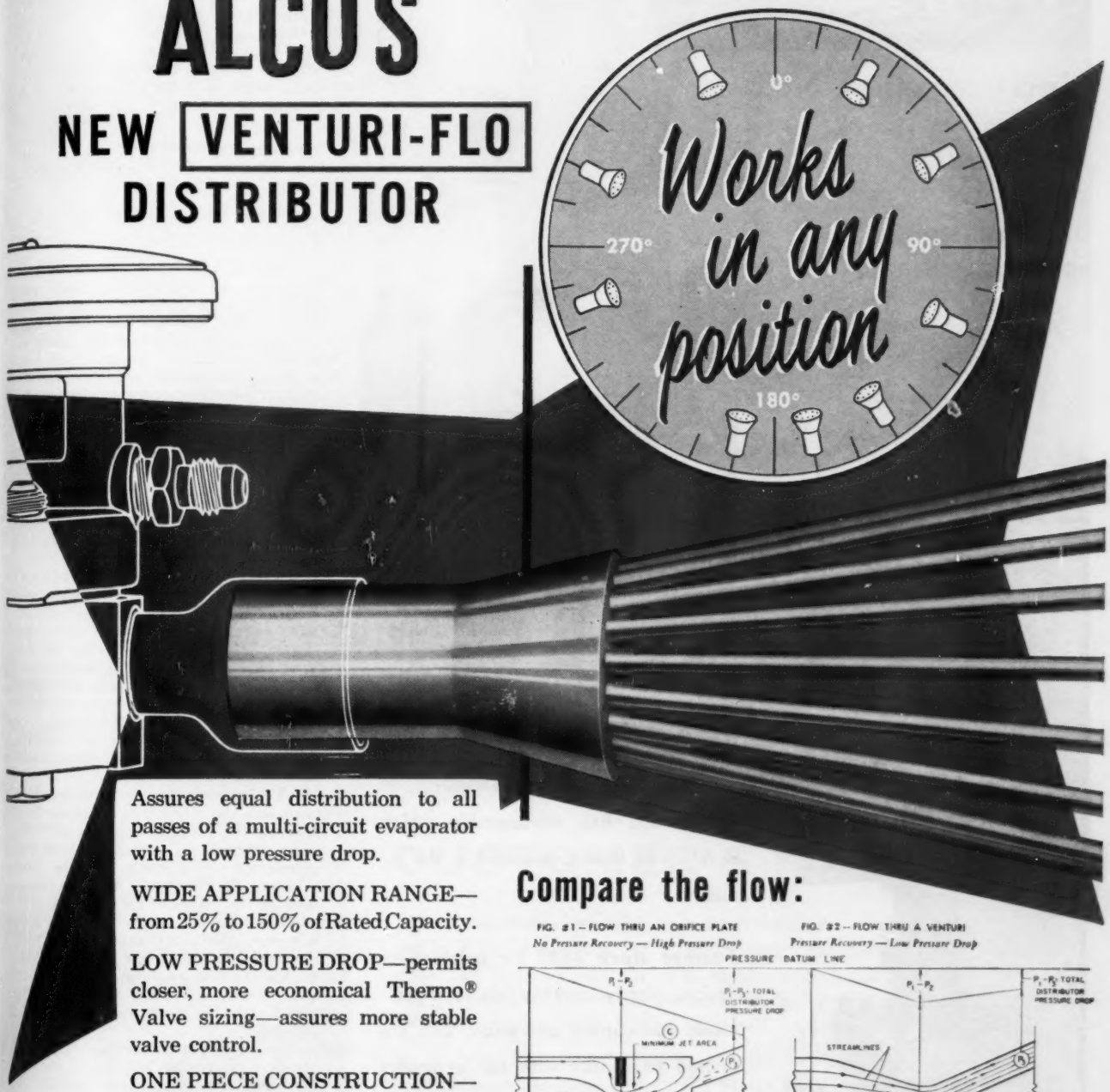

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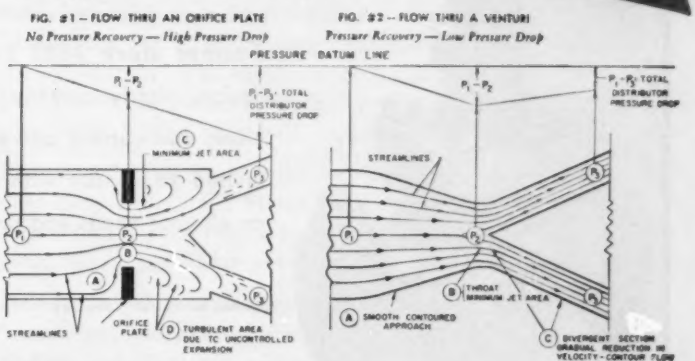
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FIG. #2 – FLOW THRU A VENTURI
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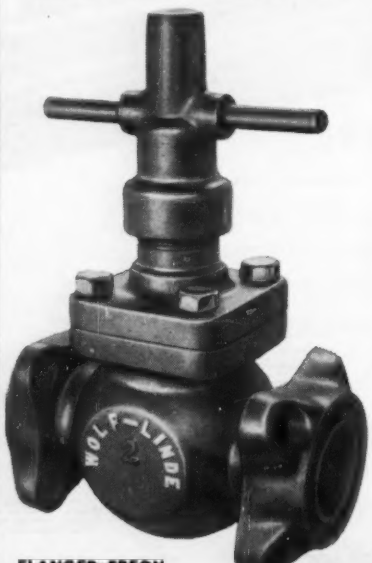

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the dehumidifying coil performance or leaving DB prediction is the non-uniform temperature of the coil surface. The fin root temperature could be much lower than entering air dew point temperature even when the average surface temperature computed by existing methods may indicate that the coil surface is dry. This is one of the reasons why conventional performance computations result in a large error when the sensible heat facter is near unity.

TAMAMI KUSUDA
Staff Engineer
Worthington Corporation

. . . SURPRISES ME A LITTLE

To the Editor:

"Predicting Leaving Dry Bulb Temperatures for Air Conditioning Coils" surprises me a little in its stress on the fact that the line indicating the change of state, on a psychrometric chart of an air vapor mixture which passes over a coil containing a boiling refrigerant at a temperature appreciably below the dew point of the vapor is a curve and not a straight line.

It seems to me, the article referred to as Reference (4) in the bibliography brings this out rather clearly even giving a method for plotting the curve which experience has proven to be reasonably accurate. The authors do not appear to be familiar with the article by William Goodman, "Dehumidification of Air with Coils" which was given at the ASRE meeting on June 24, 1936 and in which these curves are plotted for a number of cases. It has been generally known for many years that such a change is indicated by a curve unless the surface temperature of the coil does not vary. If the surface temperature of the coil is constant throughout, the change of state would be indicated by a straight line.

To say that, "for a given entering condition and a given evaporating temperature this curve is approximately the same for all coils regardless of the geometric configuration or the face velocity" only indicates that the variations in the values of these factors were too small or the accuracy of the tests was such (or both) that the effect on the curve was not apparent. Indeed, for the variation in geometric configuration and velocity normally encountered in actual use, the deviation of the curve is small.

It should not be assumed that these curves representing change of state of the mixture passing over the coil will always reach the saturation curve at the point representing the temperature of the average surface exposed to the air at the leaving side of the coil. In some cases it reaches the saturation curve at a temperature appreciably higher than that and is slightly in the fog region at the temperature of the surface on the leaving side of the coil. This is shown in the article (4) of the bibliography.

Certainly the deviation of test points from curves in Figs. 1, 2 & 3 is great enough to cover the variation caused by 400 to 600 ft/min face velocity and the different geometry of the coils tested. These variations are, in my opinion, too great to warrant any conclusion that face velocity and coil geometry do not affect the curves. I am thoroughly convinced that these factors do affect the curve and that the methods described in the article of Reference (4) in the bibliography enable plotting

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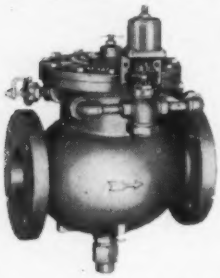

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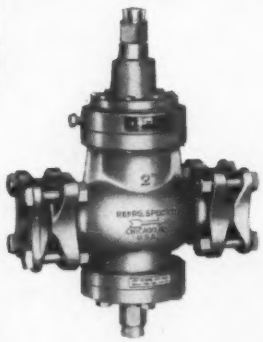

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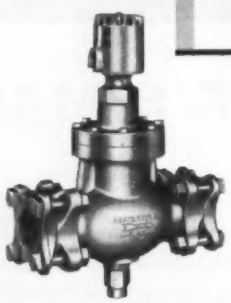

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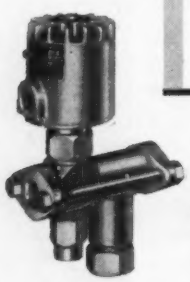

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of the approximately correct curves for a particular case if the geometry, metal and film heat transfer resistances are known. I am inclined to feel that the "K" value for sensible heat coefficients for the air side, as given in that Article (4) should be reduced about 20%.

M. A. RAMSEY
Consulting Engineer
Worthington Corporation

THAT FROST ACTION FORMULA

To the Editor:

Your correspondent, H. G. Wallace, (RE January 1959) was quite correct in assuming an error in one formula in "Frost Action Under Cold Storage Plants" (RE October 1958). This should have read

$$T(z) - T_m = \frac{(T_o - T_m)}{\left(1 + \frac{2lK_g}{r_o K_l}\right)\left(1 + \frac{2z^2}{r_o^2} - \frac{2z}{r_o}\sqrt{1 + \frac{Z^2}{r_o^2}}\right)}$$

The minus sign appearing in the denominator of the expression in the text should have been a multiplication sign.

The expression for calculating the equivalent radius r_o appearing in the text is for a slab of linear dimensions "2a" and "2b" not "a" and "b" as quoted.

D. C. PEARCE
Belmar, N. J.

HIGHLIGHTS OF THIS ISSUE

At the 65th Annual Meeting of ASHAE in Philadelphia there were 2800 attending and 20,699 registrations at the simultaneously held 14th International Heating and Air Conditioning Exposition .....p. 70

ASHRAE, off to a fast start, establishes Officer, Board of Director, General Committee and Special Committee members .....p. 102

Research physicist J. F. Harris reports upon instrumentation and methods for evaluating proposed new hermetic motor insulating materials .....p. 49

Fundamentals of chimney performance are related to laboratory tests by Professor W. G. Colborne .....p. 55

President Boling emphasizes importance of organizing ASHRAE for the future .....p. 73

Research program to expand greatly says Research Director Burgess H. Jennings .....p. 91

More emphasis on standards according to Technical Secretary A. T. Boggs, III .....p. 104

Late news highlights

Higher incomes Engineers' salaries are increasing on an average of 6½% a year, according to "Professional Income of Engineers," a report published by Engineers Joint Council (RE, January 1959, page 19). Median salary of engineers in industry, government and education in 1958 ranged from $6,125 annually for engineers with one year of experience to $11,000 for those with 20 years of experience. Despite the recession, 1958 graduates had numerous job offers and starting salaries were higher than in 1957.

A notable event With 470 exhibits, including complete heating and cooling systems and components representing various methods of heating, cooling and circulating indoor atmosphere and registration of 20,699, the 14th International Heating and Air Conditioning Exposition surpassed previous displays in size and attendance.

Nuclear powered cooling "Within ten years nuclear fission will be employed extensively to create both heat and electricity. . . . The use of such heat or electricity to drive air conditioning and refrigeration equipment should be an immediate, practical application." This is the prediction of Dr. J. F. Downie Smith, vice president in charge of Research and Development, Carrier Corporation. Although reactors can be used in this country now, they are more expensive than other energy sources; however, new designs, more experience and heavier volume will bring the cost down where it will be comparable to coal, oil and gas.

New materials studied Research on the thermoelectric properties of some materials which have thus far received little attention for thermoelectric cooling possibilities has been undertaken at Battelle Memorial Institute. Basic objective is to prepare new compounds and test their resistivity, thermal conductivity and thermoelectric power. Materials that have been investigated extensively, such as bismuth telluride, are not included in the present study, which is being sponsored by 16 industrial firms.

Planning ahead American management must rely on research and planning programs to enable it to make decisions that will keep its companies in business years from now, advised Mark W. Cresap, Jr., president of Westinghouse Electric Corporation. Management has to anticipate the changes that will take place as a result of an expanding economy and advancing technology.

Useful information A guide to sizing, installation and servicing of units, *Residential and Commercial Air Conditioning* by Charles H. Burkhardt provides information on calculating heat gains, sizing of cooling systems, installation of unit and various components and installation and operational controls. Important advances in absorption cooling, air cooled condensers, water towers and compressor and component servicing are discussed. The book contains numerous service charts and installation diagrams. (New York, McGraw-Hill Book Company, 324 pp., $9.)

Aviation meet "This Shrinking World" will be the theme of the ASME Aviation Conference to be held March 9-12 in Los Angeles. To cover developments relating to jet transports, satellites, propulsion and missile and satellite logistics, the meeting will include a session on air conditioning of aircraft.

Declining Enrollment in American engineering schools is declining for the first time in seven years, reports the American Society for Engineering Education. There were 2.9% fewer engineering students in the fall of 1958 than in 1957, and the freshman class was 11.6% smaller in 1958 than in 1957.

Old buildings air conditioned Upon completion of its reconstruction at the end of 1959, the 117-year-old Court of Appeals hall in Albany will probably be the oldest air conditioned building in the world, reports Carrier Corporation whose equipment is being used. Perhaps the second oldest is the 17-story Rockefeller Building in Cleveland, built 63 years ago and fully air conditioned by Carrier in 1957.

EJC Board Meeting

Some highlights of Engineers Joint Council Annual Board Meeting held in January were a seven point recommendation for strengthening and implementing the work of EJC; a proposal approved by the Council for steps to be taken toward eventual merging of EJC, ECPD and EMC under one corporation, with each division maintaining its own identity; and a luncheon address by Dr. Robert B. Brode, Associate Director of the National Science Foundation, on the policies of NSF toward engineering.

Pre-engineering grants

Trane Company is making available up to eight $500 two-year pre-engineering scholarships to high school seniors in the La Crosse, Wisc., area. Upon completion of the course, which is being conducted in cooperation with La Crosse State College, students may be offered positions with Trane as technicians working on engineering related projects.

Engineers and public relations

Addressing the recent Annual Meeting of the American Institute of Electrical Engineers, Dr. James R. Killian, Jr., Special Assistant to the President on Science and Technology, stated that there is a poor distinction between "science" and "engineering" in the public mind and invited the engineering profession to help bring into "more clear focus the image of the engineer." Also aware of this problem, last fall Engineers Joint Council planned a conference on public relations aspects and problems of the engineering profession, including a plan for a national advertising program to accomplish what Dr. Killian suggested, "paint for the public an accurate picture of what the engineer does." At that time it was not possible to finance the project, but it is hoped that such a program will take place in the near future.

ECPD reports

Engineers' Council for Professional Development has issued its 26th Annual Report which reviews comprehensively that organization's activities for the year ending September 30, 1958. Included are reports of all major committees.

Miniature freezer

Only 2 in. long and ⅜ in. diam., a freezing unit developed by A. J. Ross of Elmhurst, Ill., supposedly can produce 20 lb of ice per day under ASHRAE test conditions and 50 lb under ordinary household conditions. With only one special part – a small helical conveyor that removes ice flakes from the exterior – the unit is actually the complete evaporator for a refrigeration system using standard parts. Since it can be connected into ordinary refrigerator systems without overloading, the unit provides a quick means of making semi-frozen beverages.

Agenda

Among topics slated for discussion at the forthcoming annual convention of the National Association of Refrigerated Warehouses, to take place April 6-9 in Washington, D. C., are: meeting modern trends by modernization of existing facilities; refrigerated warehousing and truck transportation; effect of research on the future of refrigerated warehousing; and frontiers of public refrigerated warehousemen.

Supplement

Air Conditioning and Refrigeration Institute has issued a Supplement to the Directory of Certified Unitary Air Conditioners published in January. Listed are names of three additional manufacturers participating in the Certification Program for Unitary Air Conditioners and model ratings of four companies which did not appear in the Directory.

Still more scholarships

Educational awards totaling $20,000 will be presented to winners of a consumer essay contest sponsored by Carrier Corporation to call attention to its 1959 line or room air conditioners. A $5,000 scholarship will be awarded to winners in each of the company's four regions. While contest blanks are available at Carrier dealers, it is emphasized that no purchase is required in order to participate.

Leasing by units

Package Lease Plan for the refrigeration and air conditioning industry whereby any combination of equipment may be considered one package and leased as a unit for three to five years has been announced by Nationwide Leasing Company. This plan permits firms to acquire $10,000, $25,000, $50,000 or more in equipment without depleting their liquid working capital.

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New insulating media

for hermetic motors require new tests

JOHN F. HARRIS

CARLO B. SONNINO

Primary purpose of the project discussed in this paper was to develop laboratory instrumentation which would make it possible to evaluate new types of insulating material electrically, chemically and physically. At the inception of this work it was considered likely that the apparatus finally developed would not resemble closely any commercially available devices. Many of these had been developed to satisfy conventional testing procedures which did not reveal much about the fundamental nature and properties of insulating materials. When an interest was shown in determining possible uses within our organization for new insulating materials, it was decided that a new basic physical approach was necessary.

It was intended that only those testing procedures which proved genuinely useful in determining the suitability of insulating materials for specific applications would be retained. Unnecessary testing methods, regardless of how much they had been sanctified by traditional use, were to be discarded. The program was to be kept as flexible as possible because with the number of new insulating materials that are now available, there is probably no single set of testing procedures that will evaluate all of them accurately and fairly.

John F. Harris, Research Physicist, and Dr. Carlo B. Sonnino are with Emerson Electric Mfg. Co. This paper was presented as "Instrumentation and Methods for Evaluating Proposed New Hermetic Motor Insulating Materials" at the 45th Semiannual Meeting of ASRE in New Orleans, La., December 1-3, 1958.

Developing test instrumentation is not an end in itself. There must be a purposeful and carefully thought out philosophy behind any extensive instrumentation-development program. The basic aim of the present program is suggested by a question asked by the chief electrical engineer of one of our organization's customers. "Recently I removed a hermetic motor from a compressor that has been operating for seven years. The rag base paper slot liners were oil soaked; the cuffing on the slot liners was frayed; the wire enamel had begun to deteriorate in places; the motor in general was a disreputable looking sight. However the motor performed satisfactorily and if it had been left in the compressor, it might have given several more years of good service. Should this motor be considered a good motor or a bad motor?" As this question suggests, the primary purpose of a testing program is to determine if the material or appliance will perform satisfactorily the function for which it was intended.

The insulation evaluation program resulted in the development of a number of new test instruments. The instruments discussed in this paper include: (1) a controlled ambient chamber for hermetic use, (2) a high potential wire film continuity tester, (3) a device for determining the cut-through temperature of wire coatings and insulating varnish films, (4) a modified type of Elsee tube, (5) high temperature and pressure stability test bombs, (6) a hermetic Soxhlet extraction bomb for refrigerant extractions, (7) improved blister test bombs and (8) an all purpose Nema wire twist fixture.

The qualifying dielectric tests, cut-through tests, comparison surge tests, etc. for the components of the insulation systems of hermetic motors are now made in air at room temperature and pressure. This is unrealistic because hermetic motors operate in an ambient of oil and refrigerant gas at various temperatures and pressures. A better correlation can be obtained between laboratory test data and actual operating conditions if the qualifying tests are conducted in a chamber in which the ambient conditions encountered in a hermetic compressor shell are simulated. Such a procedure is essential when investigations are made of new insulating materials. A controlled ambient

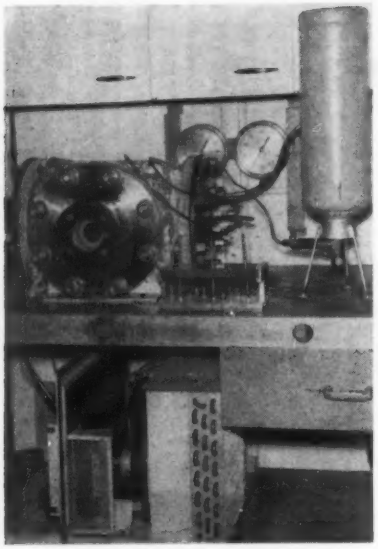


Fig. 1 Controlled ambient chamber

chamber was built in the Research Department to fill this need. The basic shell of the Controlled Ambient Chamber consists of a T-section of cast steel pipe with an 8-in. inside diam. The three access openings to this pressure vessel are closed with ½-in. thick boiler plate covers bolted to the pipe flanges. Teflon covered gaskets assure a pressure and vacuum seal. The unit has an ASME pressure vessel rating of 125 psig. A ⅜-in. diam copper tubing wound around the outside of the T-section and acting as the evaporator of an external refrigeration system cools the device to temperatures that are found in the suction side of a compressor. Some important features of the Controlled Ambient Chamber are:

1. The chamber is connected directly to a manifold through which it is possible to evacuate the chamber to pressures as low as 110 micron or to charge it to any desired pressure with refrigerant gases.
2. 10 specially designed spark plugs in one of the cover plates make it possible to introduce electric power at heavy currents and low voltages or low currents with voltages as high as 9000, whichever may be desired for the particular test being conducted. By changing the external connections to the spark plugs it is possible to conduct several tests in succession without opening the chamber. Other special spark plugs with copper and constantan electrodes are also used to connect pyrometers to thermocouples inside the chamber.
3. High pressure Pyrex windows are provided in two cover plates so that phenomena occurring in the chamber during tests may be seen or photographed with a rapid-print camera which can be attached to a special mounting on one of the windows.
4. Provisions have been made to mount a compressor inside the chamber. An external evaporator and condenser will complete the test refrigeration system. This compressor accepts 55 frame hermetic motors and the stators may be changed readily during testing operations. It is possible to observe and photograph the motor winding during the operation of the compressor.
5. Provisions have also been made to connect a small high pressure, high temperature vessel to the Controlled Ambient Chamber with a hermetically sealed solenoid operated valve opening and closing the connection between the two chambers according to a predetermined cycle. Metal, metallic oxide, and organic contaminants will be placed in the small high temperature, high pressure vessel and allowed to react with the refrigerant ambient. During the portion of the cycle when the valve is opened, the contaminated gas will flow back into the larger chamber which duplicates conditions in the suction side of a compressor.

Herewith are listed some of the test procedures now studied in the Controlled Ambient Chamber.

1. **Cut-through test for wire enamel.** A small fixture has been built which makes it possible to perform cut-through tests on 3 samples of magnet wire in sequence without disturbing the system seal. The only other known facility where cut-through tests may be made in a refrigerant ambient is at the research laboratories of one of the copper companies. A special type of cut-through test for determining the surface resistance of insulating varnish films on copper shim stock may also be made with the present equipment.
2. **Effects of corona on insulation materials.** When the pressure of the refrigerant gas in the ambient chamber is reduced to values between 120 and 300 microns, test samples may be totally immersed in an extensive but controlled corona glow. Usually a 5000 volt ignition transformer with adjustable, controlled input is used to supply voltage to the specimen under test. Infrared spectrographs, made before and after test, can determine any chemical degradation that has occurred either in the insulating material or in the refrigerant gas ambient.
3. **Burn out system contaminants.** At present it is possible to clean a refrigerating system in which a motor failure has occurred by purging with the refrigerant. Certain film insulations, rag base paper and plastics impregnating varnish leave no by-products that cannot be removed readily. Manufacturers of refrigerating

WIRE CONTINUITY TEST

Magnet Wire Coating		No. Of Punctures Per 100 ft.	Length Tested	Peak Test Voltage
Formvar A	Min.	0	2000 ft.	7.7 KV
	Mean	3		
	Max.	6		
Formvar B	Min.	0	2000 ft.	7.7 KV
	Mean	4		
	Max.	6		
Acrylic A	Min.	10	500 ft.	
	Mean	15		
	Max.	20		7.7 KV
Hi Temp Polyurethane A	Min.	30	500 ft.	
	Mean	35		
	Max.	44		7.7 KV
Hi Temp Polyurethane B	Min.	15	500 ft.	
	Mean	20		
	Max.	35		7.7 KV

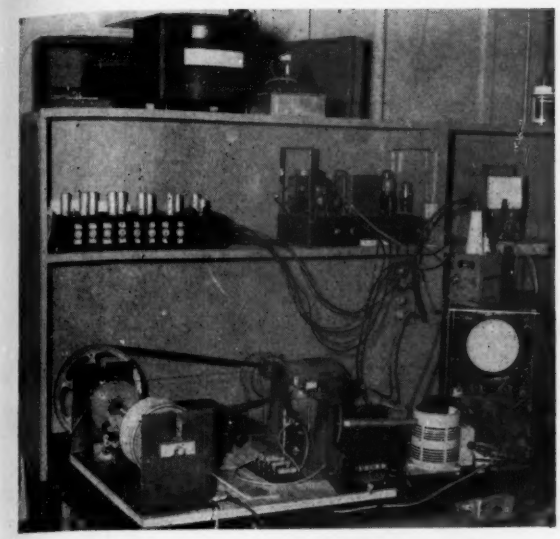


Fig. 2 Wire film continuity tester

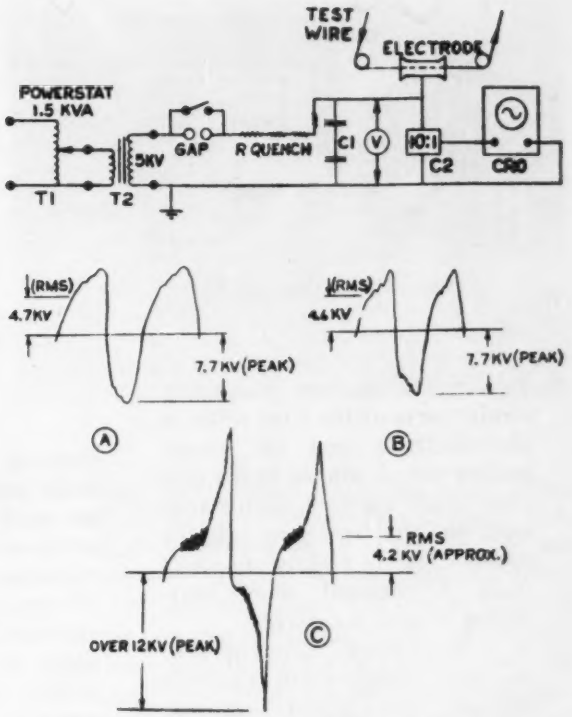


Fig. 3 Continuity tester (schematic diagram with wire counter and detector circuits omitted). (A) Output with transformer T2 and leads alone; (B) with electrode, test wire, C1 and C2; (C) "Fault" oscillogram

equipment are deeply concerned, however, that some of the new proposed insulating materials may have combustion residues that cannot be removed with conventional safe solvents, such as the refrigerants. If this is the case, an entire system might have to be scrapped after a motor failure. In the Controlled Ambient Chamber 55 frame hermetic motors may be burned out under actual operating conditions and the auxiliary tubing evaporator and condenser may then be studied to determine the character of the burn out contamination.

4. **Refrigerant gas decomposition.** The decomposition of refrigerant gas under simulated operating conditions may be studied in two ways. With the ambient chamber kept at compressor suction side conditions, a tungsten filament is heated to represent a wire burn out. The refrigerant gas is then analyzed for chloride or fluoride ion content. In the alternate method contaminants are introduced into the high temperature, high pressure auxiliary vessel and allowed to react with the refrigerant gas. The gas in the main chamber is then examined for decomposition.
5. **Dielectric tests.** All standard dielectric tests, corona starting tests, and repetitive comparison surge tests can be conducted on insulation systems in the Controlled Ambient Chamber.

Use of the Controlled Ambient Chamber has opened a relatively new field in the investigation of hermetic insulation materials. Tests conducted in this chamber could bridge the gap existing between physical and electrical laboratory tests and accelerated life tests made in customers' refrigerating systems.

Development work has been completed on the non-destructive Continuity Tester for magnet wire insulation films. This device seems to be superior to other non-destructive devices now available for detecting crazing and imperfections in magnet wire films. The Continuity Tester is a high potential continuity tester. As shown in the schematic of Fig. 3 a reel of wire to be tested is guided into a cylindrical electrode with flared ends and pulled at an average speed of 15 fpm. This relatively slow speed is believed better for this laboratory application than the 100 fpm production test speeds. A moderately peaked voltage is applied to the circular electrode. With the test wire always at "ground", pinholes, and other types of discontinuities of the enamel are detected by means of the resultant arcs that form when the breaks arrive at the electrode. A "take up" reel rewinds the tested portion of the wire. A counter indicates total wire length reeled out, and another registers the arc count. The test results are in terms of number of faults (arcs) per 100 ft. It is non-destructive as proved on "twist test", (before and after test).

The Continuity Tester simply provides a uniform ionizing field for detecting physical flaws of the enamel in question by causing an arc to strike for each flaw. In order to permit proper testing conditions, however, the equipment must provide: (1) constant alignment of the wire with respect to the electrode, (2) constant tension, (3) constant voltage, (4) constant waveform and (5) spark quenching. Some details of these will be mentioned here.

1. Parallelism and centering of the wire is accomplished by rigid insulated guides on each side of the high-voltage electrode. The closer these guides are to the electrode, the better the control on the wire. Limiting separation—they must not be so close as to cause corona between the holders and electrode.

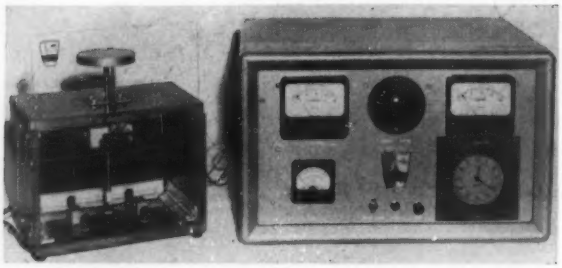


Fig. 4 Cut-through tester.

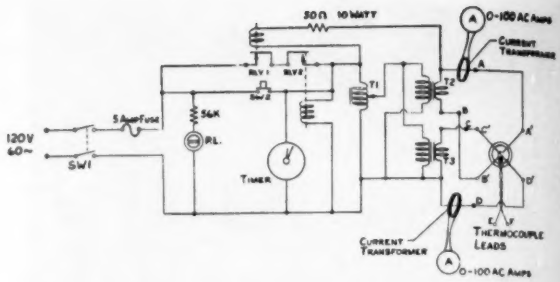


Fig. 5 Cut-through tester (circuit diagram). T2 and T3 to give 75 amp each through A′-B′ and C′-D′. T1 autotransformer, 1.0 kva

2. Tension is desired to insure straightness of the wire while in the electrode, and for proper reeling out. A simple brake provides fairly constant sliding friction on the test reel at the nearly constant reeling-out speed used. A constant speed gear-motor is used for drive.
3. & 4. Parameters involved are: electrode configuration; electrode voltage, and shunt capacity. These include length and diameter relationships for the wire size to be tested and for the peaked-ness of the electrode voltage. With the ignition transformer available for the project (a Webster 110; 5000 volt rms 125 volt-amp transformer) the wave form of the no load secondary voltage remains essentially the same from 95 to 125 volt primary with some increase in rms and peak values.

   A flared cylindrical electrode shape was chosen after trying out different sizes of toroidal shaped electrodes. Additional shunt capacitance was added because it appeared that at "no fault" condition a certain level of ionization was maintained. The limiting value for shunt capacitance is about 1000 mmf.
5. Spark quenching is realized by three means: current limiting resistors, the low short-circuit characteristic of the transformer (high reactance type) and the eventual separation of a fault from the electrode (as wire travels out).
6. Voltage control is achieved by means of an auto-transformer.
7. This device is completely automatic and counts the number of failures per 100 ft. of wire.

A new type of insulation cut-through tester for magnet wire has the following features:

1. Two samples of magnet wire crossing each other at a 90° angle are pressed down against an anvil of silver solder which contains an iron-constantan thermocouple. The silver solder anvils are thermally insulated from the rest of the device by a ceramic tube. Heat is generated within the magnet wire specimens by passing controlled currents, varying from 35 to 45 amps through both wires. All heat is provided by these internal I^2R losses.
2. When cut-through occurs a current sensitive relay stops the test when the current passing from one wire sample to another reaches the value of 0.5 amp. A potential of 115 volt, + or — 0.1 volt, is maintained between the two wire specimens during the test.
3. During the test a dial indicator (reading to 0.0001 in.) may be used to record the magnitude of the thermoplastic flow of the wire enamel.
4. With this device it is possible to obtain curves showing thermoplastic flow, wire temperature, and current plotted against time. Such curves provide a better understanding of the nature of some of the new synthetic materials offered as insulation to the electrical industry.

The insulation cut-through tester may also be used to determine the cut-through temperature of insulating varnish films. Strips of copper shim stock five mils in thickness and ¼ in. wide are coated with the varnish to be tested. The copper strip is heated by a heavy current which passes through it and the temperature of the strip is sensed by a thermocouple in a silver solder anvil under the center of it. A ball point pen (without ink) is pressed into the varnish film by a calibrated weight. When a current varying between 0.4 and 0.6 amp. passes from the ball point pen element to the copper shim stock, the test is terminated and the time of puncture is recorded.

The Carius tube type of sta-

CUT-THROUGH TEST RESULTS*

Samples	Cut-through Temperature	Heating Current	Test Time
#8134	375 F	45A.	3 min.
Terebec F	440 F	50A.	1' 40"
#100/356/1A Polyurethane	360 F	45A.	2' 30"
High Temp. Polyurethane	410 F	45A.	5 min.
High Temp. Heavy Formvar	330 F	45A.	1' 30"
#60 Polyurethane	395 F	45A.	3 min.
#62 Polyurethane	395 F	45A.	3 min.
#64 Terebec F	455 F	50A.	2 min.
#55 Polyurethane	360 F	45A.	2' 30"
#66 Polyurethane	360 F	45A.	2' 30"
New Formvar	300 F	45A.	0' 45"
Acrylic A	410 F	45A.	5 min.
Acrylic B	390 F	45A.	3 min.

* A and B are code letters for suppliers. All wire coatings are 18 AWG heavy.

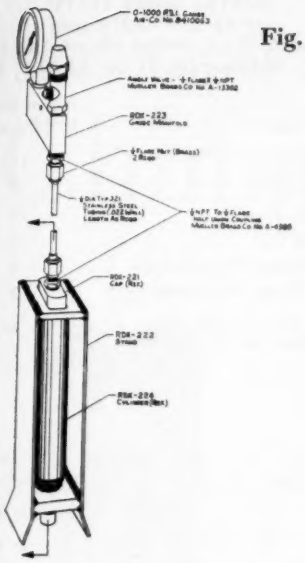


Fig. 6 Stability test bomb

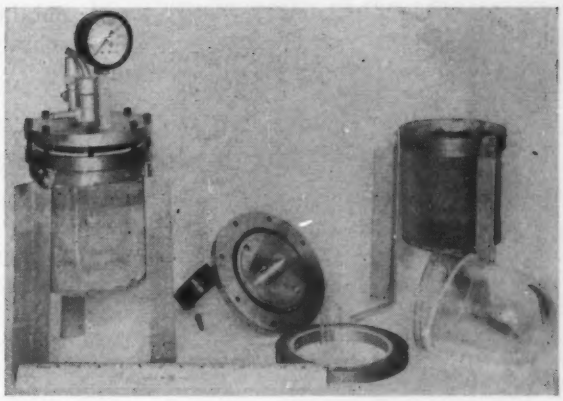


Fig. 7 Refrigerant extration bomb, assembled and disassembled

bility test in which material or materials are placed in a sealed glass vessel and subjected to high temperatures and pressures to determine pressure and temperature stability and to study compatibility has certain disadvantages. Preparation of the glass specimen tube requires great care and the percentage of breakage is quite high. Also, it is impossible to record pressures, and without a precise knowledge of pressure variations the value of the test is diminished. The Elsee tubes used by Dr. Kvalnes of duPont in investigating the stability of refrigerants are Carius tubes in miniature and suffer from the same disadvantages.

In our research department a small steel bomb of special alloy has been developed which makes it possible to study all the Carius tube reactions without danger of glass breakage and with an accurate recording of temperatures and pressures. Twelve of these bombs have been built and are now used to study the stability of hermetic insulating materials in refrigerant and oil ambients.

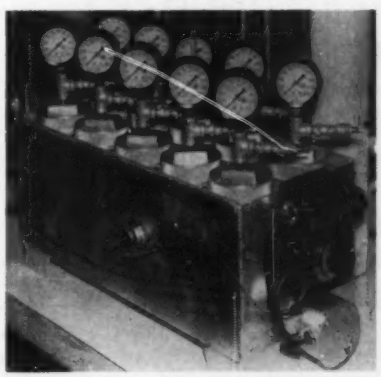
The interior of the stability test bomb is effectively 5-7/16 in. long and 25/32 in. diam. The bomb is made from 1¼ in. diam. Stellite Hastalloy B Bar. The seal between the bomb and bomb cap is obtained with an O-ring. A process valve in the bomb cap is used for charging and gage connections. A 2 point, 7 day recorder is used to obtain a printed record of variations of bomb temperatures and pressures.

The general construction of these bombs (See Fig. 6) is such that a dozen can be accommodated simultaneously in a medium-sized laboratory oven without difficulty.

These bombs, made from material as inert and contamination free as the glass in the Carius tubes, increase greatly the number of stability tests that can be conducted by a small laboratory staff. When pressures as well as temperatures are recorded during stability tests and these data are coupled with infrared analyses of the materials made before and after the stability tests, knowledge of the new insulating materials will be extended.

Although the determination of methanol and trichlorethylene extractables is quite important in qualifying a hermetic type of insulating material, it is still more important to determine what extractables will be taken out of the material by the various refrigerants. Extraction bombs were built in our laboratory which were based on work done by Harry B. Bolsen of the General Electric Company. This type of bomb is really a hermetically sealed Soxhlet extractor.

Fig. 8 Blister test bombs and reservoir



In a sealed explosion proof glass globe is suspended a small self-siphoning glass container for the extraction. The stainless steel cover for the glass globe has in it a water cooled condenser. A 150 watt incandescent bulb placed under the apparatus provides the heat source.

The material which is to be tested for extractables is placed in the self-siphoning cup and after the bomb is evacuated, it is filled with about 8 oz. liquid refrigerant. When the extraction cycle is started, the heat from the incandescent bulb causes the refrigerant to vaporize. As the vaporized refrigerant condenses at the top of the bomb it drips into the siphoning cup. When the siphoning cup is filled to the proper level, the liquid siphons off automatically and the extraction cycle begins again. Each extraction cycle takes 10 to 14 min.

The specimen is weighed carefully before and after extraction on a Microgramatic Balance. The extraction process sometimes results in a gain in weight for the specimen. This is due to the fact that the refrigerant enters into solid solution with the insulating material and cannot be removed even by baking at 200F for a half hour. It is necessary to remove the residue from the bottom of the large out-

Fig. 9 Blister test bomb disassembled

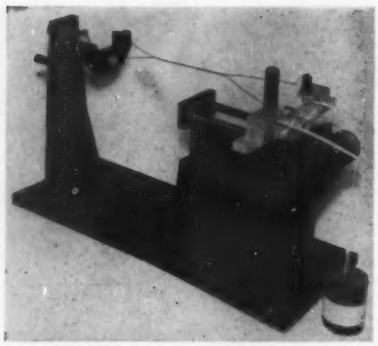


Fig. 10 NEMA twist test fixture

side glass globe with spectrographic grade chloroform and to separate the extractables by distillation. The use of this technique has made possible precise determinations of refrigerant extractables in hermetic insulating material. Some of these data may be found in the accompanying tables.

To improve the technique for testing the blistering effects of the hermetic ambient on magnet wire coatings, blister test bombs of steel were built which could be immersed in an oil bath whose temperature was thermostatically controlled. The bombs are provided with pressure gages and when they are filled, after evacuation, with 50 cc. of Suniso 3G oil and 2 oz. of Refrigerant 22 and maintained at a temperature of 40C, the bomb pressures do not vary more than 3 psig from bomb to bomb. The bombs are designed so that they can be unloaded in 10 to 14 sec and the results obtained are more exact than those obtained with other procedures.

CRAZE-FREE SAMPLES

Realistically, the proper preparation of Nema magnet wire twist samples has been a great concern to our research department. It was discovered that any handling or forming of the Nema twist after it has been prepared tends to cause crazing at the Y shaped ends of the twisted pairs. With some wire enamels this crazing is audible if the Nema twist is manipulated close to the ear. It was decided to build a Nema twist fixture on which the magnet wire twist could be prepared and tested without being touched by human hands. When this fixture was first used, it was discovered that the dielectric equipment available in the laboratory would not provide high enough ac and dc voltages to puncture the insulation in many cases. Even the comparison surge test voltages available were not sufficiently high. The adoption of this new Nema twist testing stand has necessitated a redesign of our dielectric testing equipment. The main advantage of this device is that it is now possible to determine the effects of twisting and tensioning on magnet wire film without introducing handling variables. Tests are more consistent and reproduceable.

As testing is not an end in itself it may be asked if after all the development work on new instrumentation any conclusions were reached about the virtues of the many new synthetic materials proposed for hermetic use. In this connection it should be pointed out that the laboratory tests which have been discussed are only a stepping stone towards the final and decisive evaluation tests, namely, accelerated life tests in refrigeration systems and field tests. However, it may be stated with appropriate reservations that some of the newer modified polyurethanes show the greatest promise as wire coatings and insulating varnishes. Some questions have been raised, however, about suggested polyester film slot lining materials. Dacron seems to be satisfactory tying cord and lead wire insulating material. The main advantage derived from the instrumentation development was the ability to make more rapid and careful evaluations of hermetic insulating materials.

BLISTER BOMB TESTS*
at 40 C, 190-210 psig
with 50 cc. Suniso 3G oil, 2 oz. Refrigerant 22 for 16 hr.

Wire Enamel	No. of Blisters in 4 ft Length
Formvar A	15
Formvar B	0
Acrylic A	10
Acrylic C	0
High Temp. Polyurethane A	4
High Temp. Polyurethane B	2

* A, B & C are code letters for suppliers. All wire coatings are 18 AWG heavy.

STABILITY BOMB TESTS At 310 F, 100-150 psig in Gas for 72 hr.

Atmosphere	% Weight Change			% Thickness Change			Loss of Flexibility			Color Change		
	Polyester Film	Rag Base Paper	Modified Cellulose	Polyester Film	Rag Base Paper	Modified Cellulose	Polyester Film	Rag Base Paper	Modified Cellulose	Polyester Film	Rag Base Paper	Modified Cellulose
Gas — Vacuum	−0.024	−0.29	−4.51	+2.5	Material Thickness Too Variable for Close Measurement	Material Thickness Too Variable for Close Measurement	None	S	M	None	To Brown	Lighter
Gas — Helium & Argon	−0.18	−3.71	−3.4	+2.0			None	M	SL	None	To Beige	Lighter
Gas — Refrigerant 22	+0.48	−5.81	−3.12	+3.0			None	S	SL	None	To Med. Brown	To Brown
Gas — 0.3 cc. Water & Refrigerant 22	——	−4.27	−3.96	—			S	S	SL	To White	To Med. Brown	None
Gas — 1 cc. Water	——	−0.91	−3.4	—			S	SL	S	To White	To Blue Gray	To Brown
Gas & Oil — Hlium & Argon & Oil	+0.78	+1.34	−.55	+3.0			M	None	SL	To Yellow	To Black Brown	Lighter
Liquid — Refrigerant 22 & Oil	+1.47	−1.43	+1.7	+2.5			M	M	SL	None	To Black Brown	None

M=Moderate SL=Slight S=Shatters

ORIGINAL COLORS — Polyester Film Colorless; Rag Base Paper Gray; Mod. Cellulose Med. Brown

A fundamental analysis of Chimney performance

W. G. COLBORNE
Member ASHRAE

W. C. MOFFATT

To the present, investigations of chimney performance have been largely of masonry chimneys of certain fixed sizes. Results gave performance which could be expected from a chimney of the same size and construction as the test chimney, but these results could not be extended readily to different types and sizes. Here is a discussion of the fundamentals of chimney performance from a theoretical viewpoint with substantiating data found from laboratory tests performed on aluminum chimneys. Empirical relations for the determination of friction loss in a chimney are presented and a general chimney performance equation is developed which may be applied to any chimney. The effects of inlet temperature, insulation, chimney height and infiltration on chimney efficiency are analyzed, and are justified by examination of the general performance equation.

Masonry chimneys have been used in most of the previous work in the study of chimney performance. As a result, a fundamental analysis leading to a general design method for aluminum chimneys has been difficult, if not impossible. This paper presents such a fundamental analysis, along with the theory on which a future design paper will be based.

Masonry chimneys were not used in this work due to high costs of construction and difficulty in changing diameters and heights. Instead, it was decided to use aluminum tubes of 4- and 6-in. diam from which chimneys of various heights could be easily fabricated. Having checked the fundamental theory, calculations could then be made which would apply to chimneys of any material and of any dimensions.

BASIC THEORY

Since the gas within a chimney is at a higher temperature than the surrounding air, it is less dense and therefore a pressure difference exists across the chimney wall at any given point. It is this difference in pressure, or draft, which causes the upward flow of the heated gas in the chimney. In an ideal chimney (one in which there is no friction and no heat loss) the draft at the thimble wall will be $(\rho_0-\rho_1)gH$. In actual practice such a condition could not exist and any calculation of actual or available draft must include an accounting of the friction and heat losses.

The loss of draft due to friction stems from two sources, (1) viscous and dynamic losses along the length of the chimney and (2) dynamic losses created by the chimney entrance. The general equation for friction loss resulting from the flow of fluids in pipes is

$$h_f = f\left(\frac{L}{D}\right)\left(\frac{V^2}{2g}\right)$$

In this expression f is a friction factor whose value depends on the relative roughness and diameter of the tube and the temperature, velocity and viscosity of the gas flowing. In the laminar region (Reynold's Number less than 2300) the value of f is given by

$$f = \frac{64}{\text{Re}}$$

For flows involving Reynold's Numbers in excess of 2300, the value of f may be found from Moody's friction chart[1] which illustrates graphically the relationship between f, Re and e/d.

Obviously the determination of an overall friction factor K for the chimney is complicated by the chimney entrance, so that the value of f determined from Moody's

[1]Exponent numerals refer to References.

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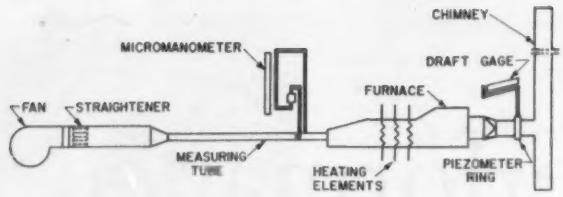


Fig. 1 Layout of chimney test equipment

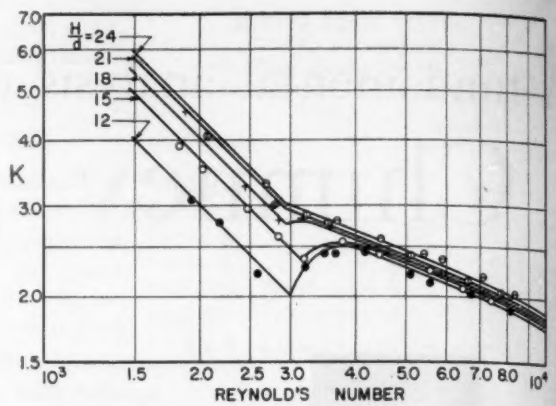


Fig. 2 Isothermal friction factor K for 4-in. diam aluminum chimneys

chart must be augmented by an additional factor to account for the dynamic losses occurring in this region. Piggott[2] found that the total loss occurring in an elbow or tee could be assumed to be made up of 2 elements, in addition to ordinary pipe friction: (1) a minimum true bend loss for smooth pipe, depending only on **r/d** where **r** is the bend radius and **d** the pipe diameter; and (2) a turbulence loss due to roughness affected by the values of Re and **e/d.** The total friction loss for a chimney expressed in terms of velocity heads would therefore be given by an expression

$$K = M\left(\frac{r}{d}\right)^{x} + N(f)^{z} + f(L/D)$$

where

M, N, x and z are constants to be determined experimentally.

L = total length of chimney and thimble under consideration.

In the aluminum chimneys, as in most actual chimneys, the entrance to the vertical flue is sharp-edged and therefore the term **r/d** may be neglected. Thus the friction loss in the chimney may be assumed to be a function of friction alone and will take the form

$$K = N(f)^{z} + f(L/D) \qquad (1)$$

The second important factor which reduces efficiency is heat loss. As the gas flows up the chimney, it loses heat to the chimney surface which in turn is cooled by the surrounding air. This results in a reduced mean chimney temperature and consequently a lower available draft. The rate of heat transfer from the hot gases to the chimney wall is dependent on the temperature difference between the gas and the wall, the area of chimney surface in contact with the gas and the inside film coefficient. The following expression[3] has been found to yield acceptable values for the film coefficient:

$$\frac{h_i d}{k} = 0.023 \left(\frac{dG}{\mu}\right)^{0.8}\left(\frac{C_p\mu}{k}\right)^{0.01}$$

where

h_i = inside film coefficient.
d = chimney diameter.
k = thermal conductivity of chimney gas.
G = mass flow of gas per unit area.
μ = absolute viscosity of chimney gas.
C_p = specific heat of chimney gas at constant pressure.

For the aluminum chimneys used in the tests outlined in this work, the thermal resistance of the walls could be neglected, and the exterior surface, being reasonably shiny, could be assumed to be non-radiating. The only other factor affecting the overall heat transfer coefficient **U** in addition to $\mathbf{h_i}$ would therefore be the outside film coefficient $\mathbf{h_o}$. It has been found[3] that for vertical heated flues, $\mathbf{h_o}$ may be given by

$$h_o = 0.29\left(\frac{T_s - T_o}{d}\right)^{0.25}$$

where

T_s and T_o = chimney surface and ambient temperatures respectively.

Thus it is possible, by integrating over the entire length of the chimney, to determine the heat loss from the chimney and from this, the chimney exit temperature. For actual chimneys, the calculation of **U** would have to be modified to account for the thermal resistance of the wall.

Appendix A outlines the development of the basic equation describing the performance of a chimney. This expression is

$$\frac{D}{\Delta\rho g H} = \frac{\Delta\rho_m}{\Delta\rho}\left(1 - \frac{K\rho_i V_i^2}{2\Delta\rho g H}\right) \qquad (2)$$

where

$D/\Delta\rho gH$ = an efficiency term relating actual draft D to ideal draft.
$\Delta\rho_m/\Delta\rho$ = a measure of efficiency drop due to heat loss.
$K\rho_i(V_i)^2/(2\Delta\rho gH)$ = the pressure drop due to friction in the chimney with isothermal flow.

A plot of $\mathbf{D/\Delta\rho gH}$ vs $\mathbf{K\rho_i V_i^2/2\Delta\rho gH}$ would therefore yield a straight line for isothermal flow*, the loss in efficiency due to friction being given by the vertical distance between the line $\mathbf{D/\Delta\rho gH = 1.0}$ and the isothermal line. Further, the drop in efficiency due to heat loss is given by the distance between the actual efficiency and the isothermal line.

Tests were carried out on the aluminum chimneys with a view to checking the theoretical equation, and noting the effects of a number of variables on chimney performance. It was considered that if these effects could be explained by the terms of the efficiency equation, then this equation could be assumed to apply to all chimneys, provided suitable accounting was made of the altered roughness and heat transfer characteristics of the chimney walls.

One variable, infiltration, while not a factor in an ideal laboratory chimney, was considered to be of sufficient significance in actual chimneys to warrant tests to study its effect. No provision was made

* Such a plot shown as Fig. 4.

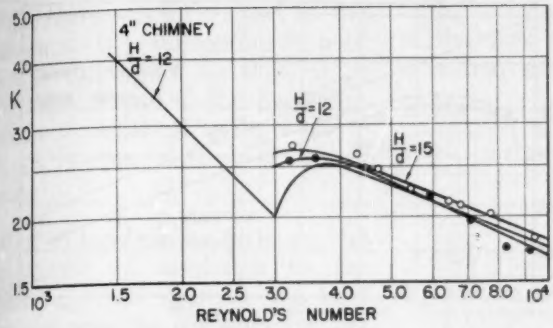


Fig. 3 Isothermal friction factor K for 6-in. diam aluminum chimneys

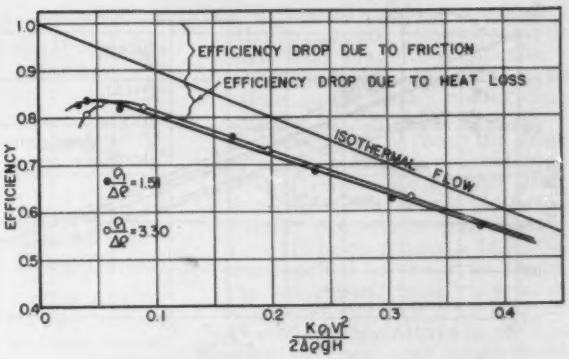


Fig. 4 Efficiency of an uninsulated 6-in. diam aluminum chimney with 2 different inlet temperatures (H/d = 15)

Fig. 5 Efficiency of an uninsulated 4-in. diam aluminum chimney with 2 different inlet temperatures (H/d = 15)

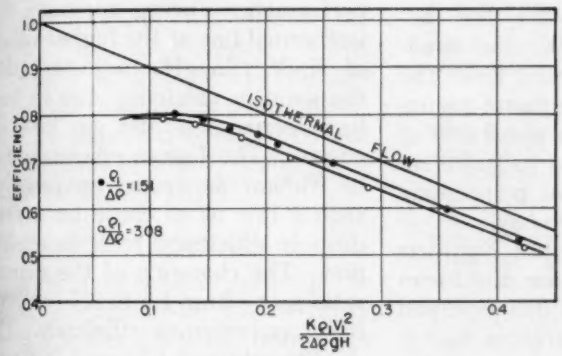
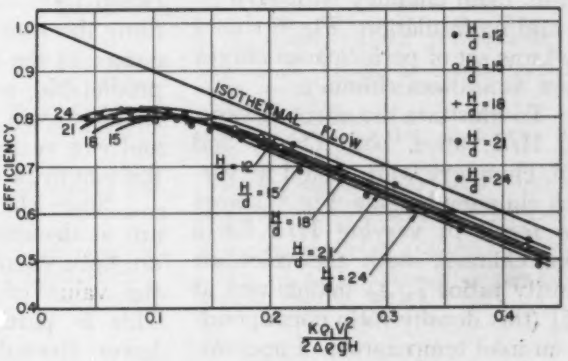


Fig. 6 Efficiency of uninsulated 4-in. diam aluminum chimneys of various heights (inlet temperature = 420 F)



for the effects of leakage in the fundamental Equation (2) but this aspect of chimney performance is of practical interest and is treated in Appendix B.

APPARATUS DESCRIBED

Fig. 1 shows the layout of test and measuring equipment. The air measuring station was located in a 1.50-in. ID copper tube, with 42 diam of tube length ahead of the measuring station to ensure fully developed velocity profiles. The air quantity was determined in the usual manner by running a 20-point Pitot tube traverse at various flow rates, from which a calibration curve was plotted. Velocity pressures were measured on a micromanometer which could be read to 0.001-in. of water.

The air was heated in an electric furnace having accurately controllable electric heaters to maintain the air entering the chimney at the desired temperature. The heated air passed from the furnace into the test chimney. The draft was measured at the piezometer ring as shown in Fig. 1; the draft gage was calibrated in position and had an accuracy of 0.001-in. of water.

Temperature measurements were made using copper-constantan thermocouples located at the chimney inlet and chimney outlet. Room temperature was measured on a shielded mercury thermometer, barometric pressure was measured using a mercury barometer, and wet-bulb temperatures were found in the usual manner. Table I gives the dimensions of the aluminum chimneys used.

TABLE I—DIMENSIONS OF ALUMINUM CHIMNEYS TESTED

Dimensions	4-In. Chimney	6-In. Chimney
Inside diameter, in.	3.90	5.86
Piezometer ring to chimney centre-line, ft	1.00	1.50
Base of chimney to thimble centre-line, ft	1.00	1.50
Piezometer ring to transition exit, in.	8.0	12.0
Tube wall thickness, in.	0.056	0.068
Height from thimble centre-line, ft	3.90 4.87 5.85 6.83 7.80—5.86 7.32	
Height to diameter ratio =H/d	12 15 18 21 24—12 15	

TEST RESULTS

K in the fundamental Equation (2) is a friction factor based on isothermal flow, defined as the head loss due to friction divided by the velocity head and depends only on Re for any given chimney. For the chimneys used in this work, K was determined experimentally with air flowing through the chimney isothermally at room temperature. The results of these tests for a 4- and a 6-in. diam chimney are shown graphically in Figs. 2 and 3 respectively.

The chimneys were then installed as shown in Fig. 1 and tested, with Reynold's Numbers at the piezometer ring ranging from approximately 1500 to 10,000. The velocity at the chimney inlet was varied from approximately 1.3 to 8.0 fps and the inlet air temperatures ranged from 230 F to 450 F.

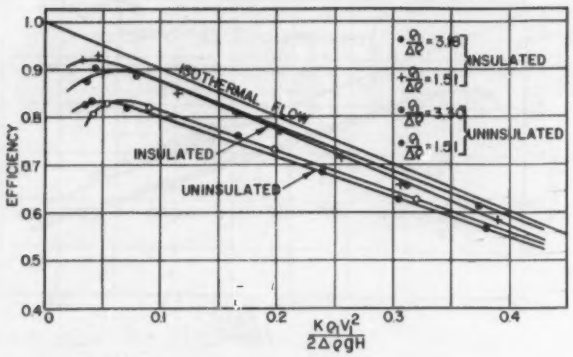


Fig. 7 Effect of insulation on the efficiency of a 6-in. diam aluminum chimney (H/d = 15)

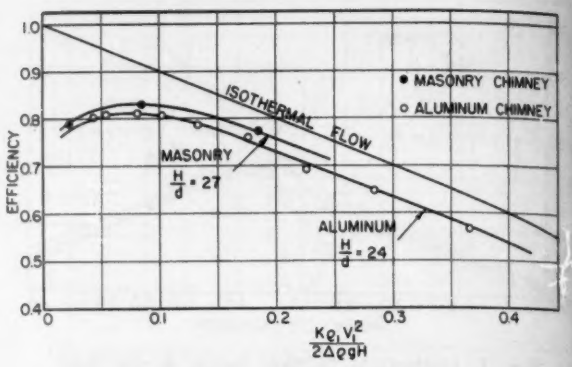


Fig. 8 Comparison of performance of masonry and uninsulated aluminum chimneys with similar inlet temperatures and H/d ratios

Fig. 4 shows a typical performance curve of a chimney using the coordinate system as just outlined. The curves shown in Fig. 4 are for a 6-in. diam chimney with **H/d** = 15 and no insulation. Fig. 5 shows the same set of performance curves for a 4-in. diam chimney.

To illustrate the effect of varying **H/d** ratios, both the 6- and 4-in. chimneys were tested at various chimney heights. Fig. 6 shows the result of varying **H/d** for a 4-in. chimney with the inlet air density ratios $\rho_1/\Delta\rho$ maintained at 1.51 (this density ratio corresponds to an inlet temperature of approximately 420 F).

While the aluminum chimneys used were essentially non-radiating, the conduction and convection heat transfer was relatively high. Realizing that a masonry chimney would have a lower heat loss than the aluminum chimneys used, it was decided to test both 6-in. chimneys when covered with a 1-in. layer of 85 percent magnesia insulation. The results of these tests are shown in Fig. 7, again for the 6-in. chimney with **H/d** = 15. The curves for the uninsulated chimney are shown as well to illustrate the difference in performance. Since the isothermal line represents the limit of efficiency attainable, it was decided that tests using a greater thickness of insulation would yield little additional information.

Some additional tests with air leakage into the chimney were performed, the results of which are outlined in Appendix B. These tests were not extensive, rather they were brief runs to evaluate qualitatively the effect of air leakage on efficiency.

RESULTS INTERPRETED

Friction Factor K: The range of Reynold's Numbers normally encountered in chimneys lies between 1500 and 10,000, thus straddling the zone of critical flow. The nature of the flow is therefore unpredictable, with the possibility of changes from laminar to turbulent and vice versa at some point along the length of the chimney.

Since the Reynold's Numbers are in the critical zone, the losses are both viscous and dynamic, and the value of K therefore varies. This is particularly true at the lower Reynold's Numbers where viscous forces predominate, rendering impossible the assumption of a constant value of K.

The values of K as determined were for the isothermal situation; in the actual chimney with heat loss the flow was not isothermal. For a constant weight flow it can be seen that Re will increase as the temperature decreases along the length of the chimney, resulting in a decreasing value of K. Obviously, it would be awkward to use a variable value for K, so that in this work, Re and hence K, were based on conditions at the chimney inlet. This is justified in most cases since the greater part of the loss due to friction takes place near the chimney entrance.

Effect of Inlet Temperature: In Fig. 4 the line of 100 percent efficiency would apply to an ideal chimney, that is, one with no friction loss and no heat transfer. The straight line marked Isothermal Flow would give the performance of a chimney having friction loss alone, while the actual performance curves as plotted take care of both friction and heat loss. It may be noticed that the actual performance curves approach the isothermal line at the higher values of $K\rho_1V_1^2/2\Delta\rho gH$, indicating that the drop in efficiency due to heat loss decreases as the gas flow increases. The loss in efficiency due to friction, however, increases at such a rate as to cause an overall drop in efficiency with increasing flow. The changing of the density ratio $\rho_1/\Delta\rho$ from 1.5 to 3.3 has little effect on chimney efficiency. The density ratios of 1.51 and 3.30 correspond to chimney inlet temperatures of approximately 420 F and 240 F respectively.

Fig. 5 shows that the efficiencies found for the 4-in. chimney are approximately the same as those found for the 6-in. chimney of Fig. 4. The only significant difference between the results shown in the 2 figures is the relative position of the density ratio curves. In Fig. 4 the curve of $\rho_1/\Delta\rho = 3.3$ is slightly above the curve of $\rho_1/\Delta\rho = 1.51$ whereas in Fig. 5 the opposite is true. Normally, it would be assumed that at a given value of $K\rho_1V_1^2/2\Delta\rho gH$ the efficiency would increase as inlet temperature decreased due to a reduction in heat loss. On analysis of the effects of T_1, T_m, $\Delta\rho$ and $\Delta\rho_m$, it can be shown from Equation (2) that the change in efficiency is dependent on the manner in which T_m varies with respect to T_1. Under certain conditions of T_1 and T_m it is possible that the efficiency will increase with an increase in T_1.

Effect of Height: The height-to-diameter ratio was varied from 12 to 24 and once again as seen from

Fig. 6 there was little effect on the efficiency. It is interesting to note that in this figure (as well as in Figs. 4, 5, 7 and 8) the maximum efficiency occurred at a $K\rho_1V_1^2/2\Delta\rho gH$ of approximately 0.1 in all cases.

Effect of Insulation: Obviously the application of insulation to the chimney would decrease the heat loss and raise the efficiency for any given value of $K\rho_1V_1^2/2\Delta\rho gH$. The extent to which the efficiency increased was of interest, however, and it was to this end that the tests on the insulated chimneys were performed. The 1-in. of 85 percent magnesia insulation was arbitrarily chosen, and would represent a greater insulation value than the normal masonry chimney. The efficiency of a masonry chimney of similar dimensions would therefore be expected to fall between the 2 curves of Fig. 7 at any given value of $K\rho_1V_1^2/2\Delta\rho gH$.

CONCLUSIONS

Main purpose of the tests on the aluminum chimneys was to verify the equations arrived at from theoretical considerations. As a result of the tests the following conclusions may be stated:

1. The general equation for chimney efficiency may be used to describe the performace of any chimney. The equation is:

$$\frac{D}{\Delta\rho gH} = \frac{\Delta\rho_m}{\Delta\rho}\left(1 - \frac{K\rho_1V_1^2}{2\Delta\rho gH}\right)$$

2. The friction drop in a chimney may be calculated provided the nature of the flow and the surface roughness inside the chimney are known. From Figs. 2 and 3 the following empirical equations may be found:

$$K = f\left(\frac{L}{d}\right) + 52\,f^{1.60} \text{ for turbulent flow Re} > 3100$$

$$K = f\left[\left(\frac{L}{d}\right) + 25\left(\frac{H}{d}\right)^{0.8}\right] \text{for laminar flow Re} < 3100$$

The K as defined by these equations includes the loss at the tee connection at the chimney entrance as well as the loss in the chimney itself.

3. The effect of temperature on chimney performance is not readily predictable. Generally, for chimneys with large H/d, increasing the centre-line chimney inlet temperature will increase the efficiency.
4. Increasing the H/d of a chimney will normally reduce the efficiency, other conditions being equal. However, the magnitude of the efficiency drop will depend to a large extent on the heat transfer characteristics of the chimney wall.
5. In order to show how the test results taken from a real chimney fit on the coordinate system used in this paper, Fig. 8 was plotted. The data of Achenbach and Cole[4] for a 7-in. ID brick chimney, 15.5 ft high were plotted on the same graph as the test results from the 4-in. diam aluminum chimney with $H/d = 24$. The inlet temperature to the aluminum chimney was approximately 420 F, while the inlet temperature to the brick chimney was 600 F. Since the brick chimney had a slightly higher insulation value, it would be expected that the efficiency would be slightly above that for an aluminum chimney. The curves show the excellent correlation which was obtained.

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Nomenclature

Symbol		Quantity	Dimensions
C_p	=	specific heat at constant pressure.	$H/M\Theta$
D	=	draft.	M/LT^2
d	=	diameter.	L
G	=	mass flow rate per unit area.	M/TL^2
g	=	acceleration due to gravity.	L/T^2
H	=	height (physical).	L
h	=	heat transfer coefficient.	$H/TL^2\Theta$
h_i	=	inside surface film coefficient.	$H/TL^2\Theta$
h_f	=	fluid friction loss in pipe.	L
k	=	thermal conductivity.	$H/LT\Theta$
L	=	total length (physical).	L
P	=	pressure (absolute).	M/LT^2
r	=	radius of pipe elbow.	L
ρ, $\Delta\rho$	=	density, density difference.	M/L^3
T	=	temperature.	Θ
T_m	=	temperature of chimney gas in an isothermal flow situation which would give the same chimney performance as in the actual case where the gas enters at T_1 and leaves at T_2.	Θ
U	=	overall heat transfer coefficient.	$H/TL^2\Theta$
μ	=	absolute viscosity.	M/LT
V	=	velocity.	L/T
V	=	kinematic viscosity.	L^2/T
x, z	=	experimental exponent constants.	O
Z	=	height (physical) above datum.	L
M, N	=	experimental constant factors.	O

Dimensionless Groups

K	=	Euler's number = isothermal friction factor.
f	=	Darcy's friction factor $= \frac{2gdh_f}{LV^2}$.
Re	=	Reynold's Number $= dV/\nu$.
e/d	=	relative roughness of pipe.

Subscripts

1	=	chimney inlet.
2	=	chimney outlet.
m	=	mean.
o	=	ambient.
S	=	surface.

Dimensions

The M L T Θ H system of dimensions was used in this paper. The symbols and units are as follows:

M	=	mass in pounds.
L	=	length in feet.
T	=	time in seconds.
Θ	=	temperature in Fahrenheit degrees.
H	=	heat in Btu.

APPENDIX A

If it is first assumed that there is no heat loss between points 1 and 2 of the chimney (Fig. A-1), it is apparent from Bernoulli's equation that

$$SP_1 + VP_1 + Z_1 = SP_2 + VP_2 + Z_2 + \text{losses}$$

where

SP and VP = static and velocity pressures respectively.

Z = a height above a given datum.

Now $(Z_2 - Z_1) = H$ and since the flow is isothermal, $VP_1 = VP_2$.

Therefore,

$$h_1 = h_2 + H + \text{friction losses} \qquad \text{(A-1)}$$

where

h_1 and h_2 = static pressures in feet of chimney gas at points 1 and 2, respectively.

Expressing the friction losses as $\frac{KV_1^2}{2g}$, and converting to pressure units, equation A-1 becomes

$$(h_1 - h_2)\,\rho_1 g = \left(H + \frac{KV_1^2}{2g}\right)$$

$$\rho_1 g = P_1 - P_2 \qquad \text{(A-2)}$$

Also, it is evident that

$$P_o = P_2 + \rho_o gH \text{ or } P_o - P_2 = \rho_o gH$$

where

P_o = ambient air pressure at the thimble level.

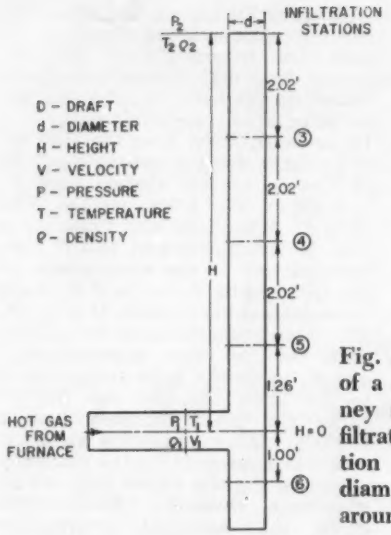


Fig. A-1 Schematic diagram of a typical aluminum chimney showing location of infiltration stations (each station consists of six ½-in. diam holes equally spaced around the chimney)

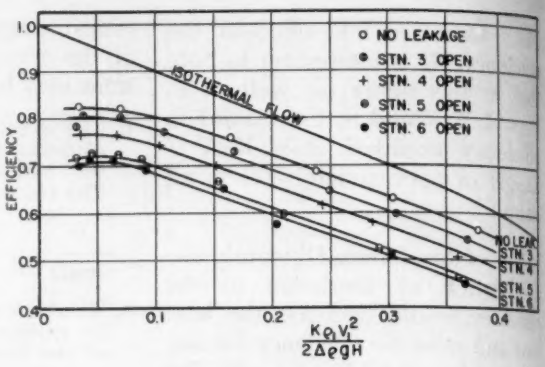


Fig. B-1 Efficiency of uninsulated 6-in. diam aluminum chimney with infiltration at various heights (T_1 = 420 F, H/d = 15)

Therefore,

$$P_o - P_1 = (\rho_o - \rho_1)gH - \frac{K\rho_1 V_1^2}{2\Delta\rho gH} \qquad \text{(A-3)}$$

Now $P_o - P_1$ represents the pressure difference between the ambient air at the thimble level and the pressure within the thimble at point 1. Hence it is the draft.

Letting $P_o - P_1 = D$, and $\rho_o - \rho_1 = \Delta\rho$, and dividing by $\Delta\rho gH$, Equation A-3 becomes

$$\frac{D}{\Delta\rho gH} = 1 - \frac{K\rho_1 V_1^2}{2\Delta\rho gH} \qquad \text{(A-4)}$$

The term $D/\Delta\rho gH$ is an efficiency term, being the ratio of actual draft to ideal draft, and $\frac{K\rho_1 V_1^2}{2\Delta\rho gH}$ is a measure of the efficiency loss due to friction.

In order to modify Equation A-4 to account for heat loss, it must first be assumed that

$$T_m = T_1 - F(T_1 - T_2),$$

where

F is a constant. T_m (with its corresponding ρ_m) is the temperature of the gas in an isothermal flow situation which would give the same chimney performance as the actual case where the gas enters at temperature T_1 and leaves at temperature T_2. For the purposes of the calculations carried out in conjunction with this work, F was assumed[6] to be 0.65. This value for F was confirmed during the experimental portion of the present work and from a paper dealing with tests performed on actual chimneys[4]. It is felt that this value ($F = 0.65$) may be used with accuracy for any chimney; however, for chimneys above 30 ft in height the value remains to be confirmed.

If one defines $\Delta\rho_m = \rho_o - \rho_m$, it is apparent from the definition of T_m that for non-isothermal flow in a chimney

$$\frac{D}{\Delta\rho_m gH} = 1 - \frac{K\rho_1 V_1^2}{2\Delta\rho gH}$$

or

$$\frac{D}{\Delta\rho gH} = \frac{\Delta\rho_m}{\Delta\rho}\left(1 - \frac{K\rho_1 V_1^2}{2\Delta\rho gH}\right)$$

APPENDIX B

A short series of tests was performed on the 6-in. chimney, $H/d = 15$, to determine qualitatively the effects of infiltration air. For the purposes of these tests, 4 infiltration stations were incorporated (stations 3, 4, 5 and 6 in Fig. A-1), each consisting of six ½-in. diam holes equally spaced around the chimney's circumference. The chimney inlet air temperature was maintained at approximately 420 F and draft and temperature measurements were made as for the normal draft tests.

The results of the tests are shown in Fig. B-1. It is immediately apparent from the curves that the lower the point at which the leakage air was introduced, the greater was the resultant drop in efficiency. This effect is readily predictable and may be attributed to 2 factors, *viz:*

1. The lower the infiltration station, the greater will be the draft at that point in the chimney, and consequently, the weight of infiltration air flowing through the holes will be greater and the resultant dilution of hot chimney gases with atmospheric air will be increased;
2. Cold air introduced at the lower stations will tend to give a lower temperature throughout the entire height of the chimney, with a greater reduction in efficiency than will the introduction of cold air near the chimney exit, which will affect but a short portion of flue.

These effects lead to the conclusion that infiltration of cold air will result in a substantial reduction in efficiency, particularly if infiltration occurs near or at the thimble level.

15,000 Gallons a Day by Freezing Salt Water to Make Fresh

Examining this model of a pilot plant for converting salt water to fresh, to be built by Carrier Corporation under a U. S. Interior Department contract, are Cloud Wampler, chairman of the board of Carrier; William Bynum, president, and Dr. J. F. Downie Smith, vice president of research and development.

Dr. Smith points to the tank in which the seawater will be frozen, converting it to a slush of pure ice crystals and brine. The brine is removed and the ice then melted to obtain fresh water. Fabrication of parts for this pilot plant, designed to produce 15,000 gallons a day, is already under way.

Five defrost methods

for commercial refrigerators

J. H. RAINWATER
Member ASHRAE

Commercial refrigerated display and storage cabinets normally use one of five defrost methods:

- Condensing unit off permitting natural defrost
- Hot gas defrost
- Electric defrost
- Water defrost
- Other external heat source defrost

The refrigerators described here are used in modern supermarkets to display and store perishables such as fresh meats, fresh vegetables, dairy products, frozen foods and ice cream. The display cases are of the open top type utilizing either a gravity air flow or forced air flow evaporator. The usual installation of this refrigerator includes a multiplicity of fixtures connected to a single remote condensing unit. In other words, the most often seen application includes several evaporators connected to one condensing unit. There are many applications employing a single refrigerator for each condensing unit. This single case is either remotely connected to its unit or is self-contained with its own built-in unit. Storage refrigerators are of the walk-in or reach-in type. The former refrigerators have one or more evaporator coils of either the gravity air flow or forced air flow type.

The temperature to be maintained in the refrigerator is one of the most important factors to be considered in the design of a defrost system. Ice cream display cases must operate with a fixture temperature between −10 F and −20 F. The refrigerant evaporates at a temperature of approximately −40 F under these conditions. Frozen food display refrigerators maintain a fixture temperature of 0 to −10 F with an evaporator temperature of −30 to −35 F. Fresh red meat refrigerators operate with a 28 to 32 F fixture temperature and an evaporator temperature of approximately 8 F. Fresh vegetable and dairy display cabinets require a temperature of 36 F to 42 F and an evaporator temperature of approximately 12 F.

The walk-in cooler or freezer must maintain a product area temperature range approximately the same as in display cases; however, the evaporating temperatures for walk-in evaporators are usually about 10 degrees higher than those for display cases due to a smaller temperature differential between the refrigerant and product area temperatures. In the design of an efficient defrost system another factor to be considered, which is closely related to the fixture temperature, is the means for disposing of the drainage. Where the drain is subjected to temperatures well below 32 F, some means must be provided to heat the drainage system.

Other factors of importance in the selection of the defrost method are the effect on the product during

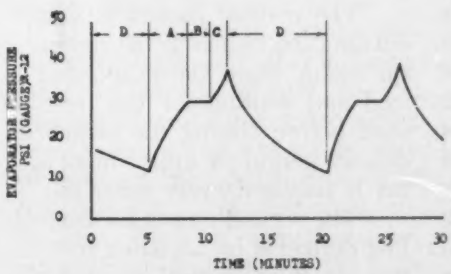


Fig. 1 Evaporator pressure graph

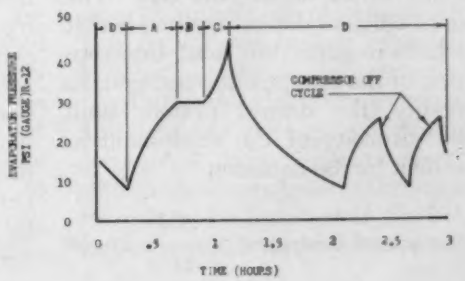


Fig. 2 Suction pressure vs. time graph for time clock defrosting

J. H. Rainwater is Director of Engineering, Warren Company, Inc. This paper was presented as "Defrosting Controls and Methods in Commercial Refrigerators" at the Commercial Refrigeration Conference at the 45th Semi-annual Meeting of ASRE in New Orleans, December 1-3, 1958.

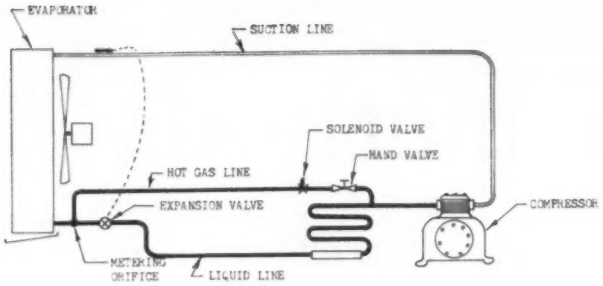


Fig. 3 Hot gas defrost using a metering orifice

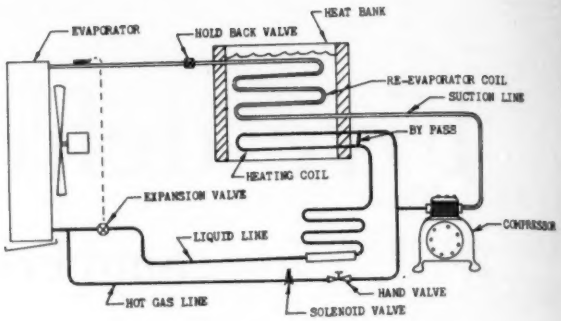


Fig. 4 Hot gas defrost using a vapor pot

the defrost period, sizing of the condensing unit, simplicity of the system and economy of the system. Certain products spoil if exposed to temperatures above their normal storage range for a prolonged period of time. Ice cream and frozen foods will thaw to an unsafe point when exposed to above 0 F temperatures for longer than one hour. Red meats begin to change to a dark color when exposed to temperatures above 36 F for longer than an hour and a half. Since some discoloration takes place during each defrost period, it is advantageous to have as few defrost periods in a day as possible. Milk stored for 48 hrs at 50 F has a bacteria growth in excess of 150 times that for milk stored at 38 F for the same period of time, thus, frequent defrost periods of prolonged duration are damaging to milk and other dairy products. Vegetables requiring less refrigeration than other products can endure higher than 38 to 42 F temperatures for long periods.

During a defrost period heat is added to the air, product fixture walls, and evaporator metal; for this reason, the condensing unit must be sized to extract this heat as quickly as possible. Some defrost systems add more heat than others in melting the same quantity of frost. Numerous defrost periods per day require that the condensing unit extract this added heat load many times during the day. The most efficient defrost method is that which requires the least interruption of the equipment operation. In reality, the defrost periods limit the capacity of the equipment as shown by the equation

$$\text{Capacity of Equipment} = \frac{Cd}{24\text{-}nt} \text{ Btu/hr.}$$

Where Cd is the daily cooling demand of the system; n is the number of defrost per day and t is the duration of each defrost in hours.

Simplicity of the system is important so that fewer installation and operating problems will arise. All factors must be carefully balanced to obtain an economically feasible system to give the desired results.

Condensing unit off permitting natural defrost is perhaps the most simple since little apparatus is involved. In this method, the condensing unit is simply permitted to remain off while the coil defrosts. Since there is no external heat added, other than that of the air surrounding the evaporator, this system is quite lengthy in the time required to melt the frost and as a result is limited to medium temperature (above 28 F) range fixtures.

Four ways to control this method are: Manual; suction pressure control (on each off cycle); time clock initiate and terminate; and time clock initiate and suction pressure terminate.

The manual control is simply opening the switch to the compressor motor when the evaporator is iced and waiting for the frost to melt before closing the switch. A defrost period of approximately 3 hrs is required every other day.

The second means for controlling defrost is by adjusting the suction pressure control so that defrosting takes place on each off cycle. This method is limited to refrigerators which can normally operate in the 38 to 42 F range. It is frequently used in vegetable and dairy refrigerators where the air through the evaporator is circulated by fans. A typical evaporator pressure graph is illustrated in Fig. 1. "D" represents the refrigeration cycle while "A", "B", and "C" represent the off cycle. "A" is the period of time during which the ice, metal of evaporator and metal of walls surrounding evaporator are rising to a temperature of 32 F. "B" is the time representing the actual melting of the ice. "C", time in which the refrigerant is rising above 32 F, is a period of most importance in allowing the defrost water to drain clear of the fixture.

"Off Cycle" defrosting has the advantage of allowing some cooling effect during the defrost period since the circulated air is flowing around melting ice. Another advantage is that the coil, free of frost most of the time, can operate near its maximum designed capacity most of the time. Even should the heat load be such that no off time can be reached, the evaporator pressure will drop as the coil becomes iced and will eventually drop to the pressure control cut-out point and allow an off cycle defrost period.

One disadvantage is that where remote condensing units are used, an ambient at the condensing unit lower than the evaporator temperature can cause the suction pressure to linger at a point below the pressure control cut-in point, keeping the compressor off for long periods of time during which the fixture temperature rises to an unsafe point. Many remote condensing units in the southern parts of the United States are installed outside and as a result during the winter months the ambient at the condensing unit is well below the evaporator temperature. The solution to this problem is to use a thermostat for controlling fixture temperature and a time clock for controlling the defrost.

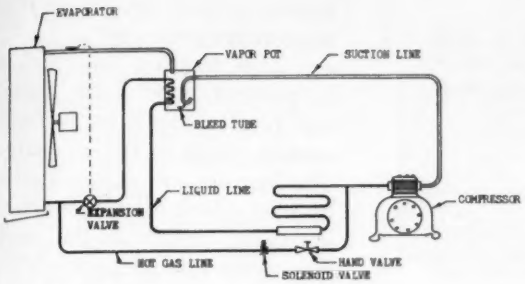


Fig. 5 Hot gas defrost using a heat bank

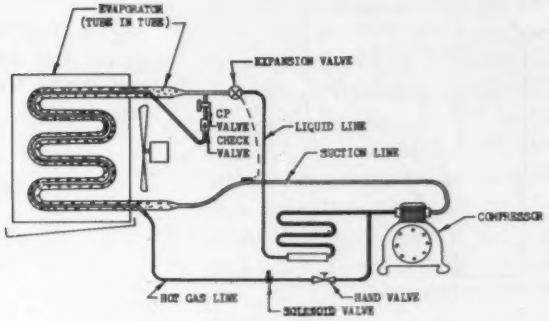


Fig. 6 Hot gas defrost using a re-evaporator

A similar disadvantage sometimes encountered is where the refrigerant lines from the fixture to the remote condensing unit are installed in trenches or conduit with numerous other cold refrigerant suction lines, which prevent the suction pressure from rising to the required cut-in points. The use of a time clock is the solution in this instance also.

The third and fourth means for controlling the defrost period both utilize time clocks. The time initiate and time terminate system merely requires that the time clock be wired to break the circuit to the condensing unit for a pre-determined time. The duration of time is normally 45 min to 90 min for forced air evaporators as used in vegetable, dairy, and meat fixtures. A period 3 hrs or more is required for gravity air flow refrigerators.

The time initiate and suction pressure terminate means of control is similar to the straight time system except that the switching mechanism for controlling the power to the condensing unit is re-energized by the rise in suction pressure to a pre-determined setting. This setting should be approximately 46 psi (gauge) for Refrigerant-12. The pressure terminate clock system has the definite advantage of allowing a complete defrost each period regardless of the quantity of frost existing on the coil and regardless of the time involved. Most pressure terminate systems have a safety time limit to terminate the defrost period by time should the pressure not rise to the pre-set point. This safety time limit is usually adjustable. The pressure terminate clock has the disadvantage listed for the suction pressure "off cycle" defrosting where low temperature ambients are experienced at the condensing unit or suction lines.

The time clock means for control is most frequently used with meat and dairy refrigerators. A typical suction pressure vs. time graph for time clock defrosting is shown in Fig. 2. This graph is representative of a meat display system where from two to four defrost periods (A, B, C) are required per 24 hrs. Dairy and vegetable display refrigerators have similar evaporator pressure characteristics and require approximately the same number of defrost periods per day. Walk-in coolers for meats, vegetables and dairy products have defrost suction pressure characteristics similar to those shown in Fig. 2. Walk-in cooler evaporators usually require one or two defrost periods per day.

Hot Gas Defrosting has many varieties all of which utilize a compressed vapor in the evaporator coil. Some systems utilize the latent heat of condensation of this compressed vapor as a heat source whereas others use only the sensible heat obtained from the highly super-heated compressed gas.

Hot gas defrosting offers several advantages over other methods. Heat is added directly to the evaporator coils without depending upon external sprays or air delivery of the heat. By applying the heat internally, a rapid defrost is obtainable. The ice sometimes becomes loose and falls off the coils before complete melting is required. To accomplish this rapid defrost, adequate drain heaters must be designed using the hot gas itself or electric heaters. Hot gas defrosting offers an inexpensive source of heat as compared to methods using other heat sources.

This method has several weaknesses that have more or less been overcome by manufacturers. The simple hot gas system merely allows the discharge gas from the compressor to by-pass the condenser through a solenoid valve and then flow directly into the evaporator coil with no metering or controlling devices in between. This simple system depends upon the compressed vapor to be condensed in the evaporator.

As the defrosting progresses the hot gas condenses in the evaporator with some remaining in the evaporator as a liquid while some goes back to the compressor to be re-circulated as a compressed vapor with the heat of compression added. As time progresses more liquid remains in the coil with less refrigerant being returned to the compressor. Since the heat source is obtained from the circulated refrigerant, the systems gradually tend to run out of heat.

Another weakness is that this system depends upon high ambient temperatures and high condenser pressures. When the condenser pressure is low the discharge gas from the compressor enters the condenser rather than the evaporator, causing an extremely low condensing pressure during the defrost which can correspond to a temperature near 32 F, resulting in little or no heat transfer from the refrigerant to the frost on the evaporator.

Danger of refrigerant liquid slug backs to the compressor is another disadvantage of this system. This liquid slug back is especially harmful just after a defrost period.

With reference to supermarket refrigerators, hot gas defrosting presents a more complex system with added refrigerant lines and controls to be installed in the field and as a result this method of de-

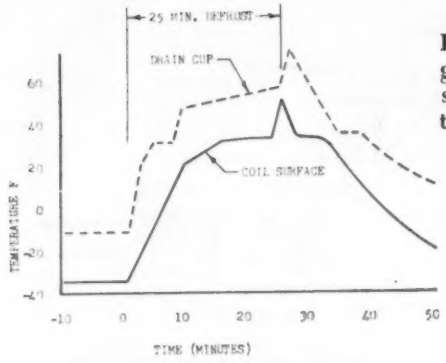


Fig. 8 Temperature vs. time graph for defrost period of self-service freezer using electric defrost

frosting is used chiefly on walk-in freezer evaporators and on self-contained frozen food display cabinets. As many as six remote refrigerated display cases are sometimes connected to one compressor which prohibits an economical use of a hot gas system.

There are several walk-in freezer evaporators available today using variations of hot gas defrosting which have overcome most of the disadvantages as mentioned above. (Figs. 3, 4, 5 and 6.)

The system shown schematically in Fig. 3 uses a metering orifice where the hot gas enters into the evaporator coil as an adiabatic (isothermal) expansion takes place. This orifice is sized to create a pressure in the evaporator lower than the condensing temperature of the refrigerant; as a result, the highly superheated vapor does not condense but gives up sensible heat to the frosted evaporator coil. Since condensation does not take place, the problem of liquid refrigerant slug back is overcome. The orifice also serves to hold the suction pressure below an excessive amount.

Controls for this system include a time clock which accomplishes three periods of time during each complete defrost period. The first interval (usually 2 to 3 min) starts the compressor and stops the evaporator fans to assure an adequate supply of hot gas. During the second period, the clock opens the hot gas line solenoid valve and energizes the electric drain pan heater. This period is usually 5 to 10 min and occurs approximately 4 times per day. The last period is a delay period during which the hot gas solenoid valve opens and evaporator fans remain stopped to allow water to drip from the coil surface and to allow the evaporator to reach a lower temperature before resuming air circulation. This period requires a setting of 3 min.

Fig. 4 shows a hot gas system using a vapor pot. This system uses an accumulator in the suction line to trap the liquid refrigerant and, by means of a bleed tube, allows small amounts of liquid refrigerant to flow back to the compressor along with the suction gas. The bleed tube is sized to overcome the liquid refrigerant slug back disadvantage but allows enough to pass to pick up the latent heat of vaporization in compression. This latent heat of vaporization is the source of heat utilized to melt the evaporator frost.

The accumulator or vapor pot has a built-in heat exchanger which is incidental and, although helpful during the refrigeration cycle, offers no help to the defrost process. It is controlled by a time clock which must have a switching mechanism designed to open the hot gas solenoid valve, stop the evaporator fans and start the compressor. (Fig. 3.) Defrost period is terminated either by the timer or, as is done sometimes, by a thermostat which measures the temperature of the evaporator and at a preset point energizes a solenoid coil in the time clock which mechanically reverses the switching mechanism to the refrigeration cycle.

The heat bank method of utilizing hot gas as a defrosting means, schematically depicted in Fig. 5, actually uses a re-evaporator coil immersed in an insulated tank of water (heat bank). The re-evaporator overcomes the refrigerant liquid slug back problem. The unusual feature of the heat bank principle is that a heating coil from the discharge line of the compressor also immersed in the tank of water builds up heat and stores it during the refrigeration cycle.

This heating coil also serves to prevent flood back during the refrigeration cycle by warming the suction gases in the re-evaporator. The heating coil is provided with a by-pass which allows the discharge gas to go directly to the condenser during the refrigeration cycle as the temperature rises in the heat bank. During the defrost period the hot gas flows to the solenoid valve, through the frosted evaporator coil, and then through the heat bank re-evaporator where heat is picked up from the hot water. The heat bank is designed with a hold back valve which maintains a low suction pressure in the re-evaporator and is set low enough to actually freeze the water. This allows the gas to not only pick up sensible heat but to pick up the latent heat of fusion given up by the freezing water. This method provides a continuous heat source not dependent upon the ambient air temperatures or condensing pressures. It also serves to hold the suction pressure below an excessive amount.

The electrical controls for the heat bank system are similar to the other systems discussed in that a timer must assure that the compressor is running, stop the evaporator fans, and open the hot gas solenoid valve. Normal defrosting with this system requires 3 periods of 10 min each which includes a 3 min drainage period.

The system shown in Fig. 6 employs an unusual re-evaporator since it is an inner tube within the evaporator coiling. During defrost the hot gas solenoid valve opens to allow the compressed vapor to flow through this inner tube, which acts as a condenser. The heat of condensation originally derived from the heat of compression is the source of heat used to melt the frost. The liquid and vapor then flow through the check valve and constant pressure valve which acts as an automatic expansion valve to maintain a refrigerant temperature above 32 F and yet introduce enough pressure drop to cause any condensed liquid to gasify before leaving the coil. The refrigerant flowing from the constant pressure valve passes into the evaporator which serves as a re-evaporator. The actual heat used for defrosting

the coil is the heat of compression of the compressor. This system is controlled with a time clock as are the other systems.

Still another system is a reverse cycle defrost method whereby a four-way solenoid valve actually reverses the flow of refrigerant during defrost to let the condenser serve as an evaporator while the evaporator acts as a condenser using the melting ice as a cooling means.

Summarizing the hot gas systems with their specific application to supermarket refrigerators, it has been established that while this method of defrosting is a good one for use on walk-in evaporators, it is too costly for use on remote display cases. The systems depicted in Figs. 3, 4 and 6 are the simplest types and overcome some of the disadvantages mentioned for the basic hot gas system. The system shown in Fig. 5 while overcoming more of the disadvantages is more complex in structure as well as in installation.

Electric Defrost, like the hot gas system, is especially applicable to low temperature refrigerators although it is frequently used in meat and dairy refrigerators to give a rapid defrost period. Electric defrost systems in most instances have heat applied externally as compared with hot gas systems where the heat is applied internally. Because of this factor and because of the limitations on the amount of electrical heat that can be applied safely, the electric defrost requires a longer interval, usually 1½ or more times that of hot gas defrost systems. However, systems have been developed which apply electric heat from within, similar to the hot gas system shown in Fig. 6, to give rapid defrost.

In any defrost system the quantity of heat required can be stated as

$$Q_d = H_F + H_A + H_S, \text{ where}$$

Q_d is the total quantity of heat required; H_F is the heat necessary to melt the frost; H_A is the heat lost to the air; and H_S is the heat necessay to warm the coil surfaces and walls to at least 34 F.

Considering first H_F, the heat necessary to melt the frost is in direct proportion to the weight of

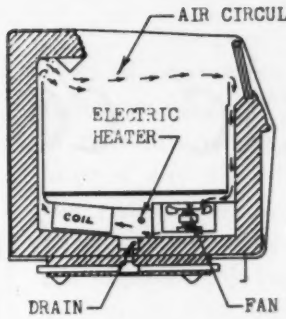


Fig. 7 Cross section of typical self-service freezer using electric defrost

frost and its temperature at the time defrost is instigated. A given volume of frost does not always contain the same weight in water. A comparatively light frost load at 20 F to 25 F would be considerably heavier than the same volume of frost at –20 F or –30 F. For an ice cream display fixture with frost at –40 F at the time defrost is instigated, H_F can be calculated to be 198 Btu required to melt 1 lb of frost and raise its temperature to 50 F as it drips from the drain. It is good practice to allow as a rule of thumb 200 Btu/lb of frost to be melted for all evaporators, both low and high temperature, as this allows a factor of safety and is an aid in the ever important post defrost drainage period. The exact weight of frost is a variable depending upon the conditions at which the evaporator is operating.

Approximate weights of frost for any given refrigerator operating under normal conditions can be obtained experimentally in the laboratory. Under normal operating conditions it has been determined experimentally that a typical 12 ft frozen food open top display cabinet employing a finned evaporator coil with forced air circulation will collect from 1 to 2 lb of frost per day. A similar display case for fresh meats would collect from 3 to 5 lb of frost per day whereas a similar forced air type dairy refrigerator would collect from 5 to 7 lb of frost. Under abnormal conditions, where the circulated air is constantly disturbed by store drafts and external air, these weights can increase as much as 100%.

The heat lost to the air, H_A, is a variable depending upon the length of time necessary to accomplish defrost and the difference in the temperature of the air blowing over the melting ice and the cold wall and product surfaces that it can see. Air having a specific heat of .0192 Btu/cu ft/OF will pick up 19.2 Btu/min for 1000 cfm circulated per degree rise in temperature. As the air rises to a temperature above that of the product and walls, it will lose the same 19.2 Btu/min for each 1000 cfm for each degree it drops in temperature as it flows over the colder product. Thus, the air returning to the evaporator during defrost can reach a point where it is colder than the air leaving the evaporator. The heat lost to the air depends upon so many variables that it is difficult to calculate its exact value. However, the values of H_F and H_S can be calculated and then arriving at a total defrost Q_d experimentally in the laboratory, the value for H_A can be isolated. For a 12 ft open self-service frozen food fixture which depends on the fans running during defrost to transfer the defrost heat to the coil surfaces, a value for H_A would be approximately 4250 Btu for a 25 min defrost period. The shorter the defrost, the less heat is lost to the air. A good design for an electric defrost system is one where the air is stopped from flowing through the evaporator by means of automatic vanes or hoods to trap the hot air.

Where the air is not flowing, other means must be provided for transferring the electric heat to the frost. This can be accomplished by inserting the heaters in the evaporator coil. However, this complicates manufacturing and field servicing problems and, as a result, in long length evaporators, it is usually best to utilize the flowing air as a heat transfer agent. Tests demonstrate that since the flowing air provides a good heat exchanger vehicle, the duration of defrost is about the same as other direct contact methods, except for systems where the heating elements are physically in the refrigerant of the evaporator.

The heat, H_S, absorbed by the coil and wall surfaces is in direct proportion to the weight of these surfaces and the total number of degrees that the metal must warm up. Using the specific heat for

(Continued on page 124)

Automatic computer

for fan testing

Normally, fans are specified and purchased in accordance with standard fan test codes[1,2] which have the approval of technical societies and fan manufacturers. These codes define properly fan characteristics and establish procedures for determining accurately fan performance, thus assuring accurate information for the design of equipment and air-handling systems.

The simplest method, which uses the conventional Pitot tube, requires a minimum of 160 pressure and power readings, and thus draws heavily on the budgeted time of a development laboratory. In design studies, which are frequently of a cut-and-try nature, progress is limited severely. Furthermore, test time cannot be reduced simply by making fewer determinations, because of the resulting loss in accuracy. Other methods, involving nozzles or orifices, are therefore often used for fan appraisal in the laboratory and prove to be sound and efficient tools. This discussion covers such a method which has been extended successfully to include rapid automatic computation and machine plotting of the fan performance curves with a high degree of accuracy.

Fig. 1 is a general view of the test chamber showing the blower mounting table and constant speed drive on the left and duct work running up to 2 exhaust blowers on the laboratory roof. Mechanical design is based on chamber specifications given in Plate XXI of Reference 3. All electrical and pressure measurements are ducted to the instrument console (shown in Fig. 2) from which speed, nozzle selection, and system resistances are controlled. Nozzles of 5, 6, and 8 in. are used individually or in combination for flow measurements. When not in use the nozzles are sealed by inflating basketball type bladders in the nozzle throat. These bladders are mounted on air supply tubing and can be inserted or retracted from the nozzle, and inflated or deflated remotely from the console.

[1]Exponent numerals refer to References.

C. H. Pountney, Jr. is Director of Research and D. W. Skipworth is Senior Research Engineer, Viking Air Products Div, National-U.S. Radiator Corp. This paper has been prepared for presentation at an early meeting of ASHRAE.

C. H. POUNTNEY, JR.

D. W. SKIPWORTH

SPEED CONTROL AND MEASUREMENT

For test purposes, fans are mounted on the chamber as shown in Fig. 3. The mounting table is positioned by an electrical drive and the fan is connected with flexible couplings through the torquemeter (strain gage transducer) to the constant speed drive as shown. The shaft coupled to the blower on the right is connected to the tachometer pickup which is wired directly to the rpm indicator located at the instrument console (Fig. 2) giving continuous speed readings. In this way, speed may be maintained within $\pm$ 1 rpm of any desired setting. Since the tachometer is on the driven side of the torquemeter, the torque used to drive it is not measured. Where physical contact is not possible with the rotating blade, such as in axial fans, speed is measured by interrupting a light beam to a photocell. This signal also appears on the speed indicator. Throughout all tests the speed is maintained constant to $\pm$ 1 rpm.

POWER CONSUMPTION

Shaft horsepower at a given speed is determined by measuring the torque input to the fan. This is done with the 0- to 200-in.-lb torquemeter shown mounted between the fan and the constant speed drive in Fig. 3. The output of the torquemeter in millivolt varies directly as the torque and thus at constant rpm the shaft horsepower does also. Since these tests are run at constant speed, the output of the torquemeter may be fed through a "weighting" network to one of the 2 pens of the function plotter to indicate shaft horsepower directly. This network is a simple voltage divider which is adjusted for different test speeds and desired horsepower ranges. The following illustrates the procedure to establish a 1.0 hp full-scale reading. At 1.0 shaft horsepower, Fig. 4 shows the torquemeter output to be 13.7 millivolt at 850 rpm. Since 5 millivolt will deflect the shaft horsepower recording pen of the function plotter to full scale, the output of the torquemeter must be reduced to 5/13.7 to establish the 0 to 1.0 hp scale. In addition this signal may be further modified to correct the data to standard air conditions by multiplying this by the ratio of standard to ambient air density. This final ratio is then set up by adjust-

ing the dial of the corresponding voltage divider.

In testing direct drive fans, watt input to the fan motor is measured with a thermal converter which produces a dc millivoltage proportional to watt input. This signal is also modified with the voltage divider network to obtain the desired span on the function plotter.

With this apparatus complete fan performance determinations are made in 20 min instead of the approximate 4 hr normally required for test and calculations. Performance at any desired speed is plotted automatically over the entire operating span of a fan in terms of brake horsepower or watt input, static pressure, and flow. Data are reproducible within ± 1 percent, and are within ± 3 percent of data obtained in conventional chambers.

STATIC PRESSURE

A static pressure tap connects the test chamber with a manometer and a 0- to 0.2-psi differential strain gage transducer in the instrument console as shown diagrammatically in Fig. 5. Both zero and standardizing controls are provided for this measuring circuit making it possible to adjust the static pressure channel in the function plotter to any desired span. The unit is standardized by introducing a simulated pressure to the transducer and adjusting the pen to the proper setting. In the present setup a calibrated resistance is applied to the static pressure transducer bridge to simulate 1.8 in. of water pressure differential. Thus, if it were desired that the full scale deflection (0 to 10) be equivalent to 2.5 in. of water, the pointer would be set, by adjustment to the bridge, at $1.8/2.5 \times 10 = 7.2$ on the scale. Data correction to standard air is made by multiplying the standard setting by the ratio of standard to ambient air densities.

AIR FLOW

Pressure differentials across the nozzles are also detected with a 0- to 0.2-psi strain gage transducer. However, because flow through the nozzle is proportional to the square root of pressure differential, it is necessary to extract the square root of the signal given by the strain gage. This is accomplished electronically with a square root extracting circuit. In addition, because there are 7 possible nozzle combinations with the 5-, 6-, and 8-in. nozzles, it is necessary to further modify the output of the square root extractor to compensate for each combination. This is done by placing the output of the square-root-taking circuit across a voltage-dividing network which has successive steps proportional to each nozzle constant. Table I shows the manner in which this is arranged.

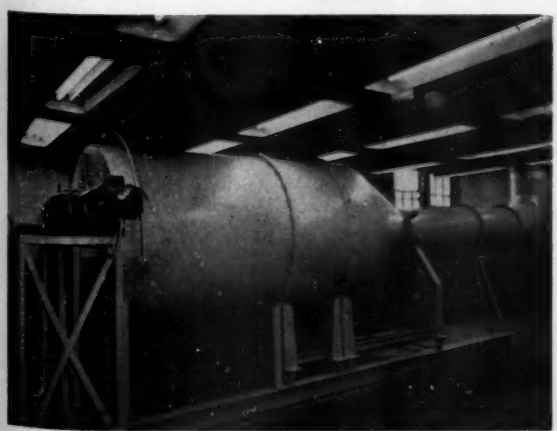
TABLE I—VALUES OF STEPPING RESISTORS IN NOZZLE SELECTOR CIRCUIT

Nozzle Combinations Diam. in.	Resistance, Ohms
5	540.5
6	778.3
5 & 6	1318.8
8	1383.7
5 & 8	1924.2
6 & 8	2162.0
5, 6 & 8	2702.5

Circuit parameters, such as the transducer and potentiometer constants, are compensated for in the process of standardizing this circuit for the desired flow span. As in the static pressure circuit, a calibrated resistor placed across the transducer bridge simulates a differential pressure by which standardization of the selected nozzle combination can be achieved. Set

Fig. 2 Instrument and control console. The 2-pen function plotter is in the upper left of the console

Fig. 1 Fan test chamber with blower mounting table and constant speed drive at the left and duct work running to exhaust blower on the right



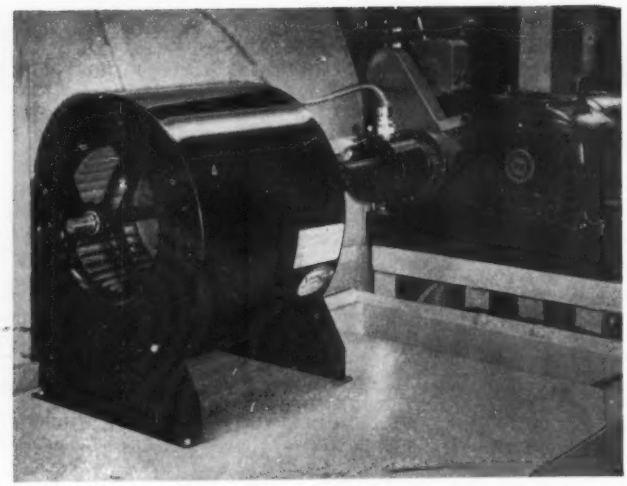


Fig. 3 Fan, as mounted on the chamber, is coupled through the torquemeter to the constant speed drive. The tachometer is attached on the driven side of the shaft

up in this manner the signal, which now represents cfm at standard conditions, actuates the drive of the chart paper in the function plotter.

COMPUTER TEST CHAMBER

Approximately 15 min is needed to mount a conventional fan on the test chamber. The fan mounting table is electrically controlled for rapid alignment with the variable speed drive and torquemeter. System resistance is controlled through a hydraulically actuated damper which modifies the flow of air from 2 exhaust blowers mounted on the laboratory roof at the end of the duct shown in Fig. 1. Nozzle selection made at the console is based on the size of unit being tested. Normally, one nozzle is sufficient to cover the entire normal operating span of a fan. If, however, accurate data at lower rates can only be obtained by changing nozzles during the test, this may be accomplished in approximately 2 min. Because the smallest nozzle presently used is 5 in., performance below 300 cfm cannot be obtained with sufficient accuracy except for static pressure at no delivery. Using all nozzles simultaneously a flow of 5000 cfm can be accurately measured.

Complete test time is approximately 1 to 2 min at one constant speed. Similar determinations at other speeds can each be made in 1 to 2 additional min. Fan performance tables can thus quickly be determined without lengthy calculation, because it is less time consuming to run a test at various speeds than to compute performance.

PERFORMANCE EVALUATED

Fig. 6 shows the performance curve of a 10-in. forward-curved fan as traced out by the function plotter. The rapid fluctuation which appears throughout the major portion of the curve represents actual fluctuations of flow conditions encountered throughout the range of performance. In transposing this curve to a master for reproduction only the mean value of amplitude is used.

The probable percentage of error introduced by the instrumentation is less than 2 percent. As a means, however, of verifying the accuracy of this equipment, results obtained with this chamber are compared with results obtained with the same fan in Plate V[4] test setup at this laboratory and at the Texas Engineering Experiment Station, College Station, Texas. Results of all 3 of these determinations are given in Fig. 7 which shows extremely close agreement between the static pressure vs flow curves. Differences between the brake horsepower curves exist because measurements made in this equipment do not include the belt friction which is the case when calibrated motors are used to determine power consumption.

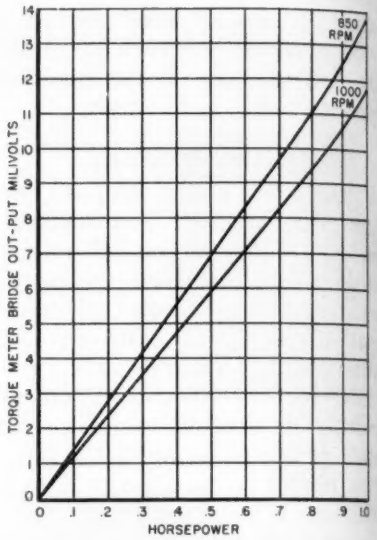


Fig. 4 Variation in torquemeter bridge millivoltage output with shaft horsepower

For rating purposes, transducer readings are checked with manometers calibrated and positioned according to Plate XXI of Reference 3 and the new proposed **Air Moving and Conditioning Association** Fan Test Code. In addition the method can be applied to a fan with an outlet duct. In this case fan performance must be established on a point-by-point determination because duct friction

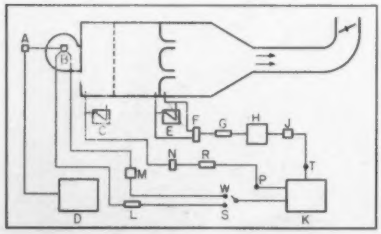


Fig. 5 Diagram of instrumentation used in computing fan performance

A Tachometer
B Torquemeter
C Manometer
D Speed indicator
E Manometer
F Transducer
G Power supply
H Square root extractor
J Nozzle compensator
K Function plotter
L Power supply
M Thermal converter
N Transducer
P Static pressure signal
R Power supply
S Shaft horsepower signal
T Air flow signal
W Watts input signal

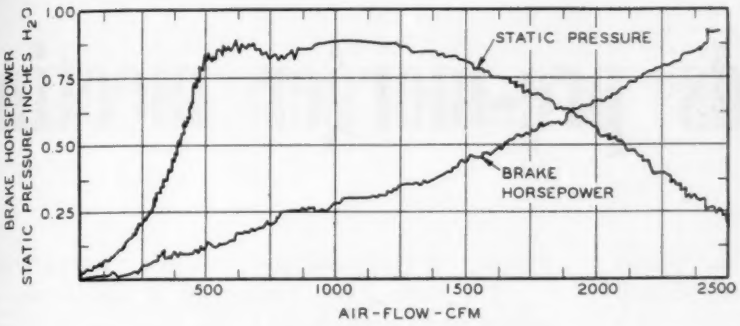


Fig. 6 Tracing of fan performance obtained with the automatic computer

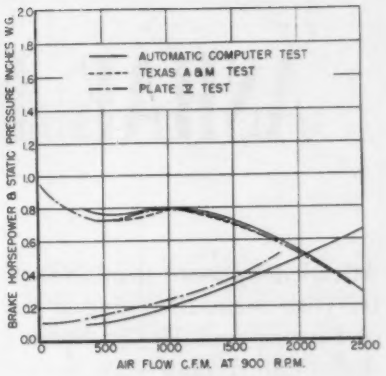


Fig. 7 Comparison of performance of a single 10-in. fan obtained with the automatic computer, a conventional Plate V chamber and a Plate V test conducted at Texas Engineering Experiment Station, College Station, Tex.

loss can be considered only at specific flow rates in the automatic calculations.

Several months' experience indicates that with routine maintenance, reliable and trouble-free operation is obtained with this instrumentation. The equipment has already proved to be a valuable tool for developmental studies and is being introduced as a means of quality control in practical fan production.

Dr. Joseph L. Hunter, professor of physics, John Carroll University, was consultant on this project and designed the nozzle selector circuitry making it possible to build this instrumentation around a chamber design which is currently in use in many laboratories.

REFERENCES

1. Standard Test Code for Centrifugal, Axial and Propeller Fans (ASHAE and the *National Association of Fan Manufacturers, NAFM Bulletin No. 110*, 1952).
2. ASME Test Code for Fans *(American Society of Mechanical Engineers,* P.T.C. 11-1946).
3. Standard Code for Testing Centrifugal, Axial and Propeller Fans (Having Wheels Less Than 12 in Diameters) *(National Association of Fan Manufacturers, NAFM Bulletin No. 116,* 1953)
4. Standard Test Code for Centrifugal, Axial and Propeller Fans *(NAFM Bulletin No. 110)* op. cit. p. 21.

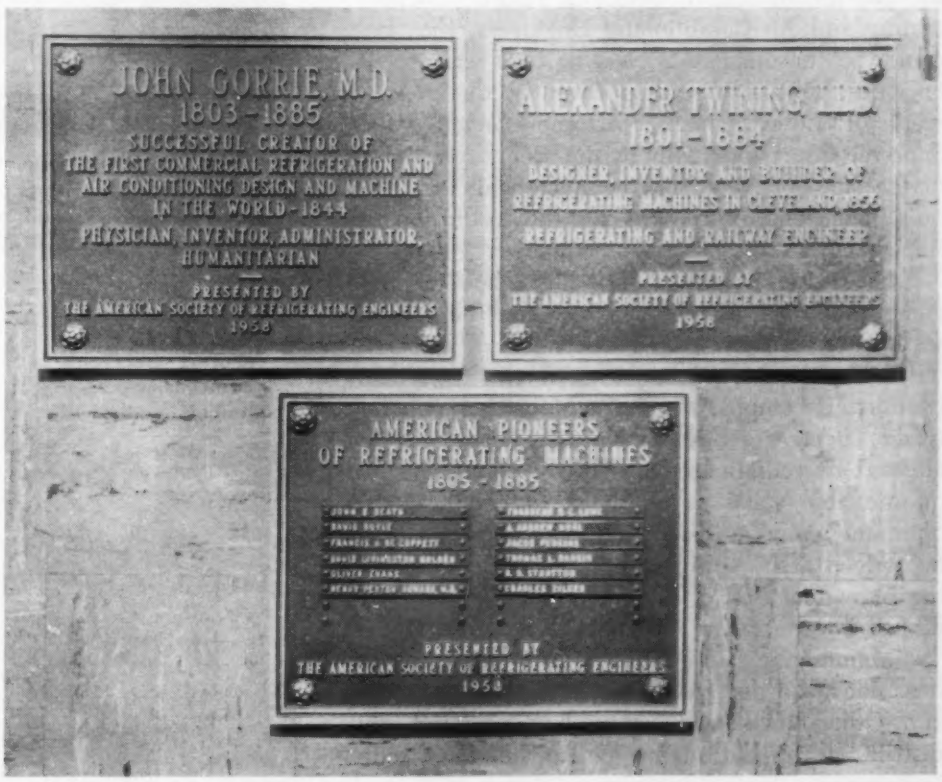
REFRIGERATING PIONEERS REMEMBERED

Authorized by ASRE Council, three bronze plaques have been completed honoring pioneers in the field of refrigeration and refrigerating equipment. These plaques will be mounted in the present uptown New York office of the American Society of Heating, Refrigerating and Air Conditioning Engineers, later to be moved to the new headquarters of ASHRAE in the Engineering Center.

John Gorrie, M.D. is cited for his creation of the first commercial refrigeration and air conditioning design and machine—in 1844.

Alexander Twining, L.L.D., is recognized for his work as a designer, inventor and builder of refrigerating machines—1856.

Other American pioneers, listed on a group plaque are: John B. Beath, David Boyle, Francis V. De Coppett, David Livingston Holden, Oliver Evans, Henry Peyton Howard, M.D., Thaddeus S. C. Lowe, J. Andrew Muhl, Jacob Perkins, Thomas L. Rankin, H.D. Stratton and Charles Zilker.



ASHAE holds pre-merger meeting

Attended by more than 2800 members and guests the 65th Annual Meeting of the American Society of Heating and Air-Conditioning Engineers was a distinguished affair – for the spirit of the occasion as reflected in the demonstrated interest and activities of those present, for the depth and character of the technical and symposium sessions and for the concurrence of the 14th International Heating and Air Conditioning Exposition. Registration at the show, where 470 exhibitors participated, was reported as 20,699.

Social highlight of the meeting was the 65th Annual Banquet addressed by Dr. Eric A. Walker, president, Pennsylvania State University, who discussed "The Morning After for Education." Honorary presentations were made to Professor Constantin P. Yaglou of Harvard University who received the F. Paul Anderson Medal for outstanding work on environmental relationships in the field of heating, ventilating and air conditioning and to retiring President Elmer Queer.

At the banquet, new officers were installed: President Arthur J. Hess, First Vice President Walter A. Grant, Second Vice President John Everetts, Jr., and Treasurer John H. Fox whose terms of office were less than 11 hours.

Indeed, it was at 9:30 A.M. Thursday, January 29, that the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., became the consolidated Society of both constituent Societies. Formal papers approved by the Department of Education, the Attorney-General, the Supreme Court and finally by the Secretary of State of the state of New York completed all legal requirements and procedures for the institution of ASHRAE.

Dr. Thomas Bedford, London School of Hygiene and Tropical Medicine, opened the topical session on comfort with a discussion of research on heating and ventilation as it relates to human comfort. He emphasized that a globe thermometer is an effective index of warmth because it is affected by radiation; that skin temperature is of questionable value as an index of comfort. An optimum temperature for a group of people can be arrived at by studying the thermal comfort of the individuals. It is desirable that thermal conditions be controlled within fairly narrow limits.

Maintaining a desired effective temperature does not mean that the environment will be pleasant. Other factors which must be considered are: distribution of heat, air movement, humidity, temperature of walls and gradients of air temperature.

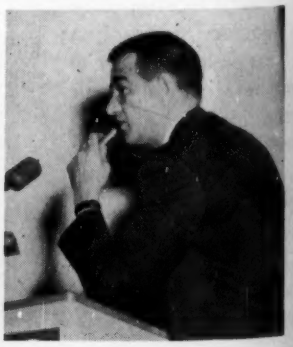
Dr. Nello Pace, University of California, reported upon evaluations made while subjects performed various activities under controlled temperature conditions and under heavy burdens of emotional and physical stress.

The opening phase of studies in the former ASHAE Research Laboratory Environment Room to reevaluate the Comfort Zone was reported by

President Elmer Queer opened the first session where the reports of the President, Treasurer and the Council were submitted. There followed a Topical Session on Comfort

Thomas Bedford cited progress in heating and ventilating as related to human factors

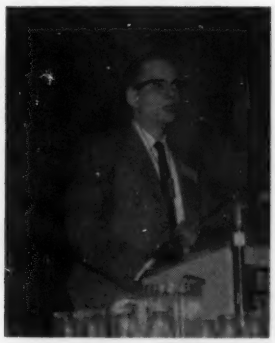
Metabolic patterns in stressful environments were the subject of Nello Pace



–it was the 65th Annual, in Philadelphia

President-for-11-hours—Arthur J. Hess completes ASHAE-ASRE consolidation and Cecil Boling becomes the first president of successor organization, the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. 14th International Heating and Air Conditioning Exposition was concurrent feature.

First Vice President A. J. Hess called the second session to order with the report of the Committee on Research as the first order of business



C. F. Speich reviewed pulsations in gas-fired residential heating equipment in collaboration with A. A. Putnam

Low frequency combustion noise in oil burners was the topic of G. J. Sanders whose co-author was W. E. Lawrie

Importance of properly locating supply outlets for basement heating was emphasized by J. R. Wright (speaking) and D. R. Bahnfleth

Burgess Jennings, Director of the Laboratory, in collaboration with Baruch Givoni. These studies were conducted in the temperature range above the Comfort Zone (80 to 105 F) with subjects in sedentary and active positions under varying degrees of high and low humidity.

J. R. Wright and D. R. Bahnfleth of the University of Illinois, at another session, discussed the results of an investigation to determine the effect of supply outlet location in a basement room on air temperature variation, floor-surface temperatures, room-air velocities and fuel consumption and to determine the cyclical variation of basement-air temperature in a system in which the thermostat was located on the first story. Ceiling diffusers, warm-air perimeter diffusers and low inside wall registers were studied.

Outlets with vertical discharge at the outside wall provided the highest level of comfort, produced the warmest floor and did not cause any localized areas of high temperature or velocity. Where outside wall locations were not feasible, low inside wall registers were recommended. Horizontal discharge outlets at ceiling level were least acceptable because of large floor-to-ceiling temperature variations and cool floors.

Room temperature control with a single thermostat was satisfactory on the basis of day-to-day changes in the temperature balance between floors. Need for zone control was evident only during mild sunny weather when the burner did not operate. Solar heat gains and internal heat loads from cooking maintained a comfortable temperature on the first story; however, basement temperatures decreased to unacceptable levels.

Increase in fuel consumption was less than would be anticipated because the basement was usually heated to some degree by vagrant heat losses from furnace casing, smokepipe and ducts. Outlets which delivered air at floor level required a greater fuel consumption than those that delivered air at ceiling level, but the space temperatures were also higher.

Basic causes of pulsations in single-port gas-fired residential heating equipment were studied by C. F. Speich and A. A. Putnam of Battelle Memorial Institute in order to obtain methods to prevent oscillations. The authors explained that oscillation involves a vortex-shedding phenomenon at the lip of the burner tube. The presence of periodic vortices passing through the flame front produces an intermittent change in the rate of heat

release of the flame. At the same time there is a pressure oscillation in the volume of the heating unit whose frequency, in the absence of driving effects, would be equal to the frequency of the unit. The degree of coupling which exists between the rate of heat release of the flame and the natural pressure oscillation governs the amplitude of oscillation produced by the unit.

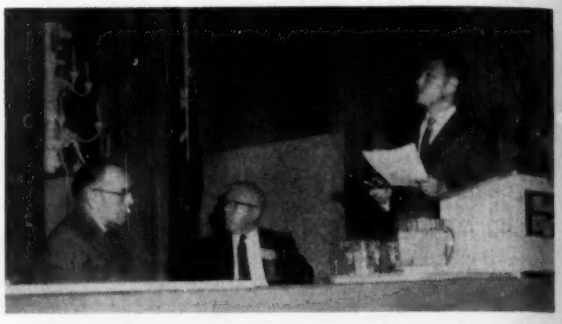
The mechanism of oscillation suggests several methods of suppressing gas-flame oscillation. Since the latter depends on the presence of a pressure fluctuation, periodic variation in the rate of heat release and the coupling between the two, any means which will change one may reduce the amplitude of oscillation. Changing normal frequency of vortex shedding; attaching a device to the lip of the burner which could either tend to restrain the vortex, cause it to shed continuously or break the large vortex into many smaller ones which will decay more rapidly; and changing the frequency of vortex shedding by changing the burner-port diameter or the velocity of the combustible gas or both, are techniques for reducing the amplitude of oscillation.

Low frequency combustion noises in oil-burning equipment are caused by random fluctuation in the burning rate of the fuel fire and the air mixture, according to G. J. Sanders and W. E. Lawrie, Armour Research Foundation. They explained that these fluctuations occur as a result of turbulent conditions in the air supply and mixing systems. Turbulence may be caused by any of these factors: fan design, housing shape, drawer assembly shape, fuel and air mixing, unstable fuel patterns or oscillatory burning due to spark ignition. Each helps to create the basic exciting force for both rumble and pulsation. The mixing process, especially in the central mixing zone, seems to be most important. In order to produce a quiet oil burner it is necessary to reduce to a minimum the creation of low frequency turbulences and to rely fully on high frequency turbulence to provide the mixing necessary for efficient combustion.

Basic rumble frequency should be increased so that it will be out of the range of resonant frequency of doors, floors and walls. A resonant frequency of 100 cps is less objectionable than the present typical frequency of 40 cps. Also, the amplification and radiation of combustion noise should be reduced by proper unit design. To prevent pulsation, the head produced by the fan must be greater than the head loss in the blast tube, the head loss in the flues and the pressure of any random pulse created in the fire box by rough combustion.

An omni-directional hot-wire anemometer for speeds below 600 fpm was described by J. F. Kemp, South African Council for Scientific and Industrial Research, in a paper presented by title.

Although the instrument described is not perfectly omni-directional, there is a fairly wide range of flow directions which do not affect its response. The hot-wire was selected as the basic type of sensing element because of its high sensitivity at low speeds and the variety of possibilities in its construction. The anemometer was spherical, of a symmetric nature, so that at any given velocity the same flow pattern and the same temperature distribution existed over the exposed surface of the

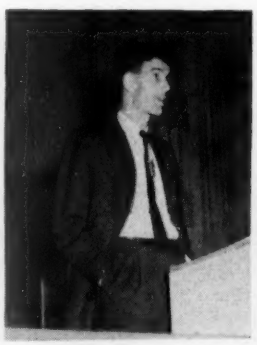


P. M. Chung sponsored several discussions of heat transfer. Here shown with A. B. Algren and Treasurer J. H. Fox

Radiant heating by hypocaust in old Roman baths and dwellings was the topic of Edwin D. Thatcher

Benjamin Gebhart reported upon a new method for calculating radiant exchanges

A probe for measuring the thermoconductivity of dry and moist materials was the subject of a paper by W. Woodside



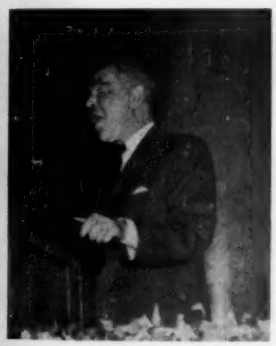
Annual use of electrical energy for space heating grows rapidly, as reported by R. L. Boyd

element, irrespective of the direction of the flow.

The constant-resistance method of measurement was adopted for the anemometer. In this method the resistance of the hot-wire element was maintained constant by adjusting the heating current, which served as a measure of the air speed.

Apart from its omni-directional properties, other noteworthy features of the instrument are its simplicity of construction and operation, the small dimensions of its velocity sensing element and a fairly quick reaction time of less than one minute.

At the Welcome Luncheon Professor Merl Baker received a Life Member plaque from E. K. Campbell as A. J. Hess and W. A. Grant observe



Feature speaker at the Welcome Luncheon was Dr. James Creese whose topic was "Independence Hall and the Kremlin"

Professor A. B. Algren receives special Life Member badge at Welcome Luncheon

Nonagenarian Charles V. Haynes, past president of ASHAE, received a rousing ovation

Heat and mass transfer processes between humid air and a solid surface kept at subfreezing temperature have become common phenomena with application of low temperature refrigeration. In a two-part study, P. M. Chung and A. B. Algren of the University of Minnesota conducted an experimental and theoretical investigation on the effect of frost formation on heat transfer between a humid air stream and a cylinder.

At the beginning of the experimental test, reported in Part I, a uniform layer of frost usually formed soon along the entire surface of the cylinder, with fastest growth near the stagnation point. In about 100 min the thickness of the layer was more or less evened out for the whole cylinder except near the sides of the cylinder, where a definite discontinuity in thickness and density of frost was observed. Mass transfer coefficient was minimal at this region for a cylinder.

Frost layer thickness reached a static value of 0.1 to 0.15 in. in about two hours along the entire cylinder surface except at the sides. In all cases heat transfer became quasi-steady in 60 to 100 min. As frost was formed on the surface the density and thermal conductivity of frost nullified the effect of increasing thickness of the frost. Increased roughness of the cylinder surface with frost formation also affected the heat transfer.

In Part II of the study, correlations were made between theoretically developed equations and experimental data. Properties of already-formed frost varied with time. The mode of variation depended on different parameters. Variation of frost density and conductivity was due largely to diffusion of vapor through the porous frost layer.

In this frost layer, vapor migration was a slow process. Heat transfer associated with vapor migration was of small magnitude compared to the heat transfer by conduction. Conductivity of frost was not appreciably influenced by diffusion of the vapor through the frost during the first 60 to 80 min of transient heat transfer period. Therefore, the time dependency of the density and conductivity of the frost during this period could be neglected.

Test results showed that a quasi-steady state in total heat transfer was reached in 60 to 80 min for cases tested. Decrease in heat transfer to the quasi-steady state was influenced largely by the Reynolds number and the sensible heat ratio for a given diameter of a cylinder. This decrease became greater as the sensible heat ratio decreased and as the Reynolds number increased.

A theoretical study of the mode of heat transfer downward through a typical joist space without conductive insulating materials was reported upon also by author P. M. Chung, but here in collabora-

tion with C. E. Lund, University of Minnesota.

In heat transfer down through a joist space, no natural convection of air was possible when the temperature varied linearly along the joist surface from the upper plate to the lower plate. Heat transfer occurred through radiation and conduction only.

However, the radiation heat exchange between the joist surfaces and other surfaces caused the temperature variation along the joint to deviate from the linear variation. This created horizontal temperature differences between the joist surface and the air. Adjacent air was then heated by the joist and rose along its surface. Upon reaching the upper plate the air stream continued its movement horizontally toward the middle of the joist space. At midway the stream met the counter stream created by the other joist; the two streams merged and descended toward the lower plate.

Equations derived from boundary layer considerations made possible calculations of heat transfer between surrounding surfaces and the air. Simultaneous solution of convection equation, rational radiation equation and conduction equation for the joist yielded an overall heat transfer through a typical joist space.

Formulas developed for predicting fuel consumption of combustion heating systems are not applicable to electric heating without excessive error, contended R. L. Boyd, Edwin L. Wiegand Co. There is a lack of properly instrumented examples of actual experience records for electrically-heated areas. In setting up estimates it was shown that personal living habits influenced costs more than weather deviations or assumptions. In commercial installations, actual energy required was far below requirements calculated for other fuels.

Use of formulas developed for combustion heating fuel requirements gives abnormally high energy requirements for heating with electricity. The reasons for these deviations should be investigated. There is a need for authentic and realistic evaluation of practical methods for predicting energy requirements for heating with electricity.

An investigation of the ancient heating method of hypocaust was conducted by Edwin Daisley Thatcher, an architect. He reported upon his study of five rooms in the Forum Baths at Ostia which were heated by this method. It was shown that the hypocaust could provide an environment comfortable to bathers in these rooms even though large window openings were unglazed. Solution rested upon the character of the predominantly radiant heating system and skillful planning in architectural adaptation to the local climate.

The Forum Baths showed that hypocaust was an efficient and successful heating method and that the ancients were aware of radiant heat both from surfaces and the sun. Further, they applied this knowledge to provide human comfort.

W. Woodside, National Research Council of Canada, discussed the probe or transient needle method for thermal conductivity. He is of the opinion that this method is satisfactory for dry materials, for results of tests on dry silica aerogel showed both reproducibility and agreement with guarded hot plate aerogel. However, results of tests on moist sawdust showed a steady decrease in measured conductivity. A test on snow indicated a variation in measured thermal conductivity during the test, while moist clay showed a steady decrease in measured conductivity.

This steady decrease in measured conductivity during tests observed in the measurements on moist sawdust and moist clay suggested a steady drying out of the material in the vicinity of the probe. It was calculated that in 10 per cent moisture clay, a layer of clay equal in thickness to the probe radius would dry completely during the 10-min test. Thus, the moisture redistribution during the test did not appear to be negligible as was commonly assumed, at least for test using thin probes.

One method to reduce the temperature gradi-

Successor to Elmer Queer for an 11-hour term of office, A. J. Hess enjoyed the novelty of being the only one to have been president of ASHAE and ASRE and to be designated as the next president of ASHRAE, following the initial term of Cecil Boling

A – C. S. Leopold was moderator of the Topical Session on Comfort. B – C. S. K. Benham, President of the Institution of Heating and Ventilating Engineers in London, addressed the 65th Annual Banquet. C – Dr. Eric A. Walker, president of Pennsylvania State University, spoke at the Banquet. D – Noise and vibration in water piping systems were explored by W. L. Rogers.

A

B

C

D

ents and moisture redistribution in probe tests is to increase the probe radius. This is contrary to the present trend in probe development which is toward smaller sizes. Since errors due to finite size of a probe may be corrected, it is advantageous to use larger probes and avoid some difficulties in testing moist materials.

Necati Ozisik and L. F. Schutrum reported upon a study carried out at the Research Laboratories on heat gain through windows shaded by metal awnings. Five awnings of conventional types and colors were studied. It was demonstrated that outside color, shape and venting characteristics of awnings are significant factors influencing heat gain. The amount of solar heat excluded by a metal awning depended on the type and color of the awning, orientation of the window and the time of day.

During the period of day when an awning prevented direct sun from falling upon the glass surface, the amount of solar heat excluded by the awning was 70 to 75 per cent on a western exposure, 75 to 80 per cent on an eastern exposure and 50 to 60 per cent on a southern exposure.

Most methods of determining the instantaneous radiant energy transfer rates between gray surfaces of an enclosure consider net transfer rates between pairs of surfaces, explained Benjamin Gebhart of Cornell University. These transfer rates are often ill-defined and awkward to use, he said. A new method for thermal radiation analysis for non-black surfaces focuses attention on total rate of radiant energy loss of a surface. In this method various radiant energy transfer rates in a room are calculated. Analysis is direct and consistent because of the way basic quantities are defined and the method is similar to the approach used for black surfaces.

Numerical operations in the calculations are lengthy, especially in the solution of equations for the absorption factors. Therefore, it is considered desirable to present absorption factors graphically in order to facilitate various types of calculations such as those for adiabatic surface temperatures.

AT THE SYMPOSIUMS

Drawing excellent attendance, the several symposiums offered a round-up of current information on the several subjects.

The Symposium on Hydronics included four papers on piping systems. B. L. Sturdevant discussed the design of piping which is used for high-

(*Continued on page* 132)

Conspicuously well-attended, the Symposium on Corrosion and Water Treatment was addressed by W. L. Andrews, Olivia Burkey, C. E. Ernst, F. S. Hodgdon and L. D. Polderman (speaking). W. G. Hole, seated next to the speaker, was moderator

Performance of water source heat pumps in the Pacific Northwest was the topic of J. D. Kroeker here shown addressing the Symposium on Heat Pump Performance. Other speakers at this session were A. D. Spillman, E. R. Ambrose, W. A. Spofford and F. L. McFadden, Jr.

To Professor Constantin P. Yaglou, for his many studies of environmental factors and relationships, went the F. Paul Anderson Award at the Annual Banquet

Second Vice President W. A. Grant called to order the Symposium on Hydronics at which P. N. Vinther was moderator

American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

So now it's ASHRAE

At 9:30 a.m. January 29, 1959 the consolidation of the American Society of Heating and Air-Conditioning Engineers (1894) and The American Society of Refrigerating Engineers (1905), as voted by members of both societies on December 1, 1958, became officially and legally effective.

The consolidated Society will preserve the values developed over the years within each of the predecessor societies and will build further within the broader possibilities inherent within the successor organization.

Here, in summary, are some of those specific things which have already emerged from early planning stages —

Fields. Engineering aspects of heating, refrigerating, air conditioning and ventilation as allied with research, development, installation, equipment, standards and industry cooperation.

Membership. Approximately 17,800 members. Initially 84 local chapters organized within 10 geographical regions.

General and Special Committees. Covering the full range of society, technical, standards and inter-society activities. See page 102 of this issue for the initial listing of General and Special Committees.

Research. Continuation and augmentation of the research program conducted at the Society's Cleveland Laboratory with supplementary grants to independent educational and investigatory institutions. (page 91)

Standards. Continuation and amplification of the Society's work on engineering standards with projection into various phases of heating, refrigerating, air conditioning and ventilation. (page 104)

Publications. The several publishing activities of the merged societies will be as follows:

ASHRAE Journal. The monthly, engineering, official publication of the Society reporting upon national meetings papers, outstanding developments reported independently, news of Chapters, engineering news of the industry and happenings related to members as well as significant developments in the component products, parts and materials useful to them in their work.

Transactions. Complete annual reporting upon the activities of the Society. Technical program, social and official business reports.

Guide and Data Books. Encyclopedic handbook presentations of authoritative information covering fundamentals, applications and equipment in the fields of heating, refrigerating, air conditioning and ventilating and issued in annual volumes upon a rotating program.

Standards. Published, as developed and approved, for distribution as requested.

Research Reports. Recording the outcome of Laboratory and sponsored projects.

Meeting Preprints. Distributed at national meetings to facilitate study and discussion of papers presented at technical sessions.

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Transient refrigeration loads

as related to energy-flow concepts

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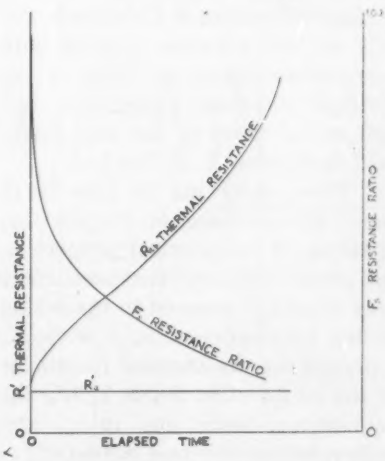
Analysis predictions of the heat-flows in a cold-storage-plant complex were undertaken, via rheo-electric simulation technique, and presented in a recent paper before ASRE.[1] Of particular interest in that initial study was the influence, on the space-temperature-time history, of a transient "hot" load suddenly introduced into the empty refrigerated zone. Of further interest was the ultimate effect of the transient warm-goods load on the performance of the different components of the entire refrigeration-complex, including the heat-transfer and the machine-equipment.

In this earlier paper, the time-effect of the warm load was handled via a concept of "transient-body resistance", that is, the warm load after introduction into the refrigerated space was represented by a *calculated* thermal resistance which was construed as varying with time. This resistance-time type of relationship is depicted by Fig. 1, similarly as shown in the earlier paper[1]. This thus accounted for a decreasing warm-body heat-flow to the storage space, commensurate with the reduction of the body-surface temperature with the passage of time. Understandingly, the warm load accounted for a time-wise disturbance of the space-temperature; this increased considerably above the design steady-state value following the introduction of the warm load. The reduction of this increased space-temperature then proceeded progressively; the energy-flow in the various components of the complex were also simultaneously affected, as, with passage of time, the warm-load heat-dissipation was absorbed.

In the present paper, the effect of transient refrigeration loads is explored further, specifically with regard to the storage-structure space and the immediately associated heat-exchanger equipment. Warm goods brought into an empty refrigerated space represents one type of transient cooling problem. However, the presence of other stored goods within the refrigerated space adds to the problem of predicting net heat-flow rates and heat-exchanger performance. Here again, orthodox calculation procedures prove inadequate, and analogy methods using rheo-electric simulation techniques are effective.

A further added development beyond the earlier paper is the *direct* experimental (via simulation) representation of the transient-body behavior, rather than by means of the hitherto somewhat indirect "transient-body resistance" technique. Calculation has been superseded by the use of a "Resistance - Capacitance" approach to establish the transient heat-flow. By specialized electrical-potential transfer procedures involving resistance-capacitance representation of both the incoming "hot" and the existent "cold" loads, the different rates of energy-flow are followed. The net energy-flow increment, beyond normal structure values, is sensed in the heat-exchanger simulation circuit; this is set up for a

Fig. 1 Typical resistance-time relationship



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specified value of brine-coolant and circulating-air flow as well as brine-supply temperature. For the integrated system, resultant space-temperature, load-body temperatures, heat-exchanger energy-flow rates and fluid temperatures are predicted as a function of elapsed time for the transient operations.

Rheo-electric analogy applications, as extensions of the resistance-concept of heat transfer, have been carried forward in a number of steady-state problems: in heat-conduction, via geometrical-sheet as well as network methods[2]; in heat-exchanger analysis[3]; in energy-transport[4]; and recently, in pipe-network fluid-flow distribution[5]. The present paper carries rheo-electrical system analysis further forward in its application to the transient refrigeration-load problem previously studied.[1] Here pilot resistance-capacitance units are used to monitor the changing heat-flows associated with the varying temperature-difference between the solid-goods surface and the storage-space air.

The general arrangement is shown in Fig. 2. The heat-load of the refrigerated space is picked up by the movement of the space-air, cooled in fan-circulation through a brine-served counter-current type of heat-exchanger. For the purpose of this study, the brine from a separate refrigeration plant is considered to be supplied at constant temperature.

Many simplifying assumptions are found to be expedient in the general development of the problem: for example, the structure, the heat-exchanger and the heat-exchanger fluids are assumed to have negligible heat-capacity; the value of the load-surface heat-transfer coefficient is assumed, and is regarded as constant in the range of operations; the same may be said for the heat-exchanger fluid-flows and their overall coefficient of heat-transfer. Likewise, for the hot and cold transient loads, values of the thermo-physical properties have been assumed and regarded as constant. The external air-temperature also remains constant.

HEAT-LOAD RELATIONSHIPS

Structure: With fresh ventilation-air (infiltration) cooling effect included, the enclosure hourly heat-

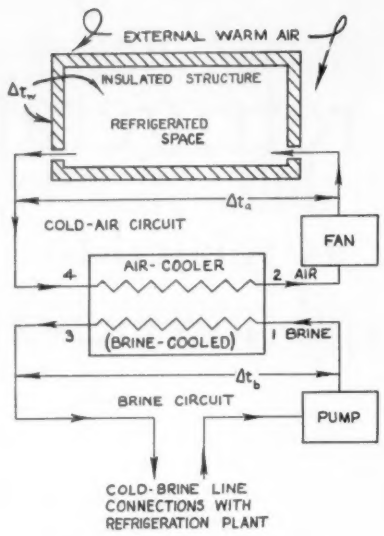


Fig. 2 General arrangement of heat flow complex

flow rate q_w, in accordance with the resistance-concept of heat transfer, may be represented by an equivalent wall-resistance R_w and the temperature-difference Δt_w between the external outdoor and the internal space-air temperature (which may be varying):

$$q_w = \Delta t_w / R_w \qquad (1)$$

Transient Load: For a solid slab-form body of given thermal properties, initially at a uniform temperature t_o throughout and then suddenly subjected to exposed-surface heat-transfer with a surrounding fluid: — the rate of heat-exchange depends not only on the exposed area of surface and its heat-transfer coefficient, but obviously, also on the realized temperature-difference existent at any time between the surface and the fluid (such as the space-air in the present case). The time-varying surface-temperature t_s depends not only on its previous history and surface-conditions as well as on internal physical properties, but also on the value of the half-thickness dimension Y of the body.

Thus, referring to Fig. 3a to depict the solid-section in question, in terms of its physical properties, and considering unit frontal-surface area (1 sq ft) exposed to the action of the air-coefficient h_a:—with R'_c representing the thermal resistance of the half-section depth Y, R'_h, the surface-resistance due to h_a, and with a as the thermal diffusivity:

$$t_s = f\left[\left(\frac{h_a Y}{k}\right), \left(\frac{(k.\ \tau)}{Y\ c\rho Y}\right)\right] = f\left[\left(\frac{h_a Y}{k}\right)\left(\frac{a\tau}{Y^2}\right)\right] \qquad (2)$$

$$t_s = f\left[\left(\frac{R'_c}{R'_h}\right), \left(\frac{\rho}{R'_c C'_s}\right)\right] \qquad (3)$$

For sustained temperature-difference between initial and uniform solid-temperature t_o and the surrounding air t_a, the dimensionless ratio F, or specifically F_x for the temperature-difference relationship at any point x for the solid (from surface to center-line core), is important:

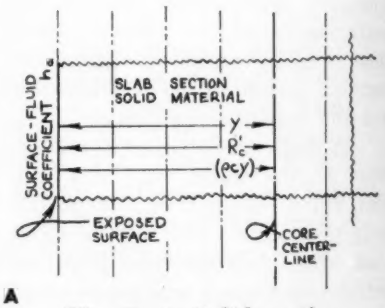
$$F_x = \left(\frac{t_x - t_a}{t_o - t_a}\right) = \frac{\Delta t_{x,a}}{\Delta t_{o,a}} \qquad (4)$$

For the surface:

$$F_s = \left(\frac{t_s - t_a}{t_o - t_a}\right) = \frac{\Delta t_{s,a}}{\Delta t_{o,a}} \qquad (5)$$

It is to be noted that Fig. 1 referred to the "transient-body" resistance R'_{tb} previously cited.[1] This may be defined as an equivalent resistance which, when divided into the overall temperature-difference $\Delta t_{o,a}$, determines the rate of heat-flow per unit of area, from which the total rate of heat-flow q_{tb} from surface of area A_s may be set up:

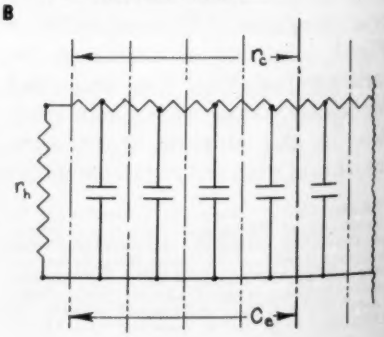
$$q_{tb} = A_s\left(\frac{t_o - t_a}{R'_{tb}}\right) \qquad (6)$$



A

Fig. 3 A—Solid-section form. B—Counterpart of A with subdivisions

B



Specifically, $R'_{tb} = \frac{1}{h_a f_s} = \frac{R'_s}{f_s}$ (7)

f_s is the specialized function for this type of case, covering the transient surface-temperature, defined by Schack[4] and originally used in the calculation of the transient-body resistance. It represents the surface temperature-difference ratio effect, that is, $(t_s - t_a)/(t_o - t_a)$, which herein is also defined as F_s. F_s may be used either as derived from calculation, or as one otherwise established, such as shown subsequently, through electrical simulation technique.

It is of interest to note a rearrangement from Equation 7:

$$f_s = \frac{R'_s}{R'_{tb}} = \left(\frac{t_s - t_a}{t_o - t_a}\right) = F_s \quad (8)$$

Parenthetically, reference should now be made to the curve for the value F_s, which has now been added to Fig. 1, and plotted against elapsed time. As a dimensionless-ratio value of resistance (Equation 8), it presents an idea which has been inspired by the freezer-development work of W. J. Finnegan here in USA. It depicts, via the ratio value F_s, the relative importance of the surface-resistance R'_s as compared to the overall, as defined through R'_{tb}; the "Finnegan" curve, in effect, draws attention to the role, often controlling, exercised by the solid-body properties, particularly in the later portion of the elapsed-time history.

The heat-flow q_L between a transient-body load and its surrounding fluid, such as air, may of course be stated in terms of the corresponding temperature - difference and the applicable resistance:

$$q_L = \frac{(t_s - t_a)}{R_L}, \text{ with } R_L = \frac{1}{A_s h_s} \quad (9)$$

HEAT-EXCHANGER RELATIONSHIPS

The air-to-brine heat-exchanger may be considered on a counter-current basis. As developed in earlier papers[1,3], its performance may be explored and predicted through an extension of the resistance-concept. Here the exchanger heat-flow q_e is related to the overall exchanger temperature - difference $\Delta t_{t,e}$ between the entering air and the entering brine $(t_a - t_b)$, through the total overall thermal resistance $R_{t,e}$:

$$q_e = \left(\frac{t_a - t_b}{R_{t,e}}\right) \quad (10)$$

$$R_{t,e} = \frac{R_b - R_a e^{(R_a - R_b)/R_{UA}}}{1 - e^{(R_a - R_b)/R_{UA}}} \quad (11)$$

with R_a and R_b representing the heat-exchange resistance of the air and brine streams, and R_{UA} representing the overall heat-transfer resistance. (e = natural log base.)

As will be apparent in the specific example cited later, proportional distance along the total resistance path $R_{t,e}$ will represent a measure of the temperature-change for the different exchanger fluids.

ELECTRICAL SIMULATION ANALOGY

Since the solid-body in transient change comprises both thermal resistance and thermal storage (heat-capacity) effects, its thermal internal behavior may be simulated electrically through electrical resistance and electrical storage effects, i.e. through resistance and capacitance. Similarly, the external surface heat-transfer effect may be represented via electrical resistance.

Thus, Fig. 3b represents the counterpart of Fig. 3a, with the surface-to-center solid section subdivided into a number of slices, each slice-center represented by its condenser, and with slice-to-slice connections (on a one-dimensional basis) via representative network-resistances, as conventionally done in unsteady-state analogies. Similar provision may be made for the surface heat-transfer effect.

As a first step in the approach, the electrical values may be considered, individually, directly proportional to the thermal values, the solid body herein being considered as the half-thickness of an infinite-extent slab (edge-effects neglected). Thus, Equations 4 and 5 have their electrical counterpart, with temperatures replaced by their voltage counterparts, and noting therefore, that in the simulation process, the dimensionless-ratio F may be determined electrically:

$$F_x = \left(\frac{E_x - E_a}{E_o - E_a}\right) = \frac{\Delta E_{x,a}}{\Delta E_{o,a}} \quad (12)$$

Thus, in effect:

$$E_s = E_a + F_s (E_o - E_a) \quad (13)$$

Or, as stated otherwise:

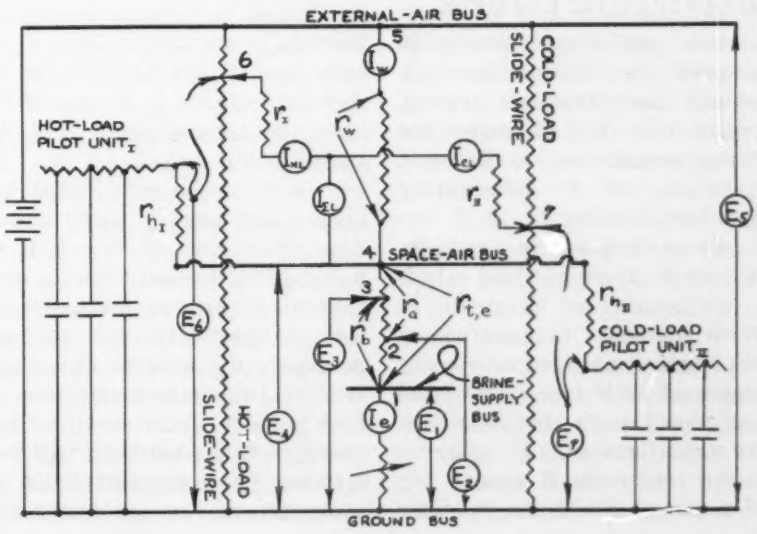
$$E_s = f\left[\left(\frac{r_c}{r_h}\right), \left(\frac{\gamma}{r_c C_c}\right)\right] \quad (14)$$

Thus, dimensionless ratio F shows as a function of the specified groups, with γ = time in sec,

$$F = \phi\left[\left(\frac{r_c}{r_h}\right), \left(\frac{\gamma}{r_c C_c}\right)\right] \quad (15)$$

It is significant that, since the electrical rC value is involved as a product, the result is independent of the value of the separate factors, and so generally understood. Therefore, for electrical simulation, the build-up of the rC product can be made in terms of convenient available values and acceptable quality of condenser-capacitance C_c, with the resistance r_c appropriately

Fig. 4 General rheo-electric simulation circuit for subsequently studied elements



evaluated. Thus, in terms of the value for r_c so chosen, and through the required value for the ratio r_c/r_h established thermally, r_h may be appropriately evaluated. In effect, this characteristic also permits a convenient choice of values so as to establish an attractive time-relationship for the "pilot" resistance-capacitance unit used (voltage-wise) to monitor the main circuit of the complex.

The behavior of this pilot-unit is then one of producing a voltage value for the "surface-temperature", which then may be separately applied in the monitoring of the main-circuit load-resistance component.

The application of this transient-body simulation approach will best be demonstrated through the illustrative example. Here, in addition to the transient problem, the concurrent behavior of the structure-load and of the heat-exchanger are incorporated into the system. Fig. 4 represents the general rheoelectric simulation circuit for the study of the subsequently-cited examples. In addition to the resistive components for the structure and for the heat-exchanger, load-transfer slidewires are also included. Pilot-unit-wise, "surface-temperature" voltages result from the operation of the rC network; their value is reproduced at the load-slidewires, so that it is thus possible to track the full-surface area load change for the main circuit (as a function of elapsed time). Thus, the flow of heat off the exposed surface of the transient (one-dimensional) loads to the surrounding air may be simulated.

ILLUSTRATIVE EXAMPLE

Time-wise performance-prediction is desired for the refrigerated-space and heat-exchanger system, represented by Fig. 2, under the following conditions: — External temperature, 100 F; brine-supply temperature (constant), 12 F; design space-temperature (steady-state), 32 F. Transient-load schedule: (a) Load x, hot, originally at uniform (outdoor) temperature of 100 F, suddenly charged into empty refrigerated 32 F space; (b) Load y, hot, plus Load x, hot, together, under conditions of (a), charged into the empty 32 F space; (c) Load x, hot at 100 F etc. suddenly charged into the 32 F space already partially occupied by Load y, cold, uniformly at 32 F.

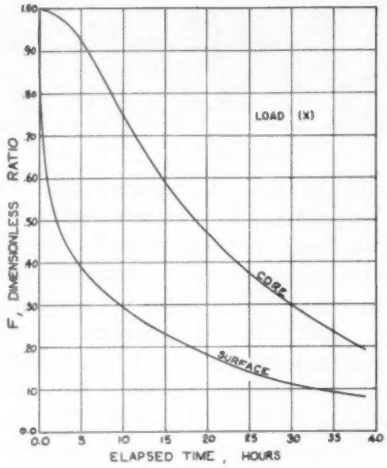


Fig. 5 Measured variation of ratio with time, load x

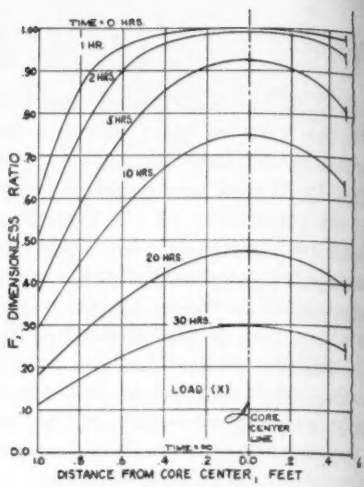


Fig. 6 Measured variation of ratio with space, load x

Thermal-load Data: Structure heat-load (ventilation - infiltration air included, and wall of negligible heat-capacity = 1260 Btu/hr-F); Heat-exchanger: brine flow-rate W_b = 8000 lb/hr, c_b = 0.70 Btu/(lb-F); air-flow W_a = 38,500 lb/hr, c_a = 0.241 Btu/(lb-F); UA = 11,850 Btu/(hr-F). Load x: Tightly-packed vertical stack of goods, 2 ft thick, of 625 sq ft total exposed area; thermal conductivity, k = 0.70 Btu/(hr-sq ft-F/ft); density, ρ = 94 lb/cu ft; c = 0.22 Btu/(lb-F); equivalent thermal diffusity, a = 0.034. Load y: Tightly-packed vertical stack of goods, 2 ft thick, of 500 sq ft total exposed area; thermal conductivity, k = 0.35; density, ρ = 94 lb/cu ft, c = 0.22; thermal diffusivity, a = 0.017. The characteristics of Load y were so chosen, to have a thermal diffusivity of one-half of Load x. For both Load x and Load y, the surface-conductance h_a is assumed to have the general value of 2 Btu/(hr-sq ft-F).

It is to be noted that both Load x and Load y, under the different programs of operation, will be subjected to heat-transfer under time-wise *variable* space-temperature. Accordingly, for reference purposes, the general characteristics of temperature-variation, for both x and y, under *sustained* heat-transfer fluid conditions will be of interest. These characteristics have been separately determined through electric-analogy resistance-capacitance studies of the "pilot loads".

Fig. 5, for the hot Load x of its defined thermal characteristics, shows the measured variation of the dimensionless-ratio F for both the exposed surface and the core, vs lapsed time, under constant-level "cooling" conditions. Fig. 6, for the same hot load, cooling, shows the variation of F vs internal position, for various value of τ, elapsed time in hours.

Figs. 7 and 8, refer to Load y, and for their defined thermal characteristics, similarly show the variation of performance for both the surface and the core, vs elapsed time, under "constant" "cooling" conditions. These are the counterparts of Figs. 5 and 6.

For these studies, and the subsequent ones on overall performance of the space system under load conditions, a time ratio of 60:1 was established for the *thermal* vs *electrical* behavior. Thus, in effect, for a value of $\tau/(R_cC_a) = a\tau/Y^2 = 1.00 = 0.034\tau$ for the hot body, giving τ a value of 29.4 hr, this represents 29.4 min = 1760 sec electrical. Similarly for the cold load, 58.8 hr thermal was represented by 58.8 min = 3520 sec electrical. Electrical circuit values were translated from the thermal numbers on a proportional basis throughout, for voltage vs temperature, for current vs heat-flow rate, and for ohms resistance vs thermal resistance.

In the voltage "transfer" technique used, based on high impedence galvanometer element, great

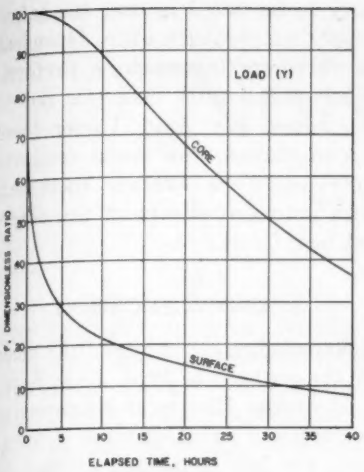


Fig. 7 Load y ratio variation with time

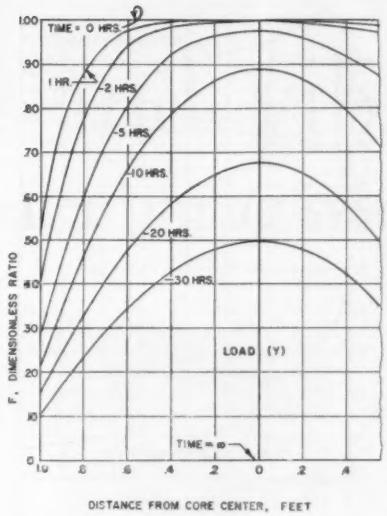


Fig. 8 Load y ratio variation with space

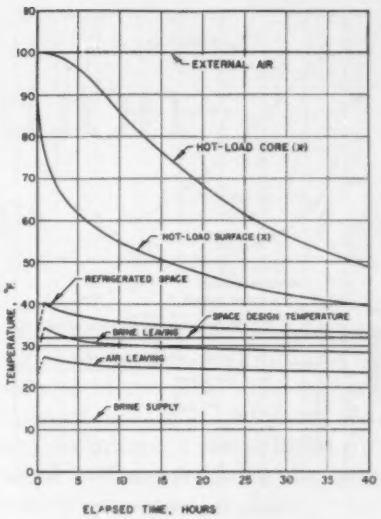


Fig. 9 Temperature-time history with hot load

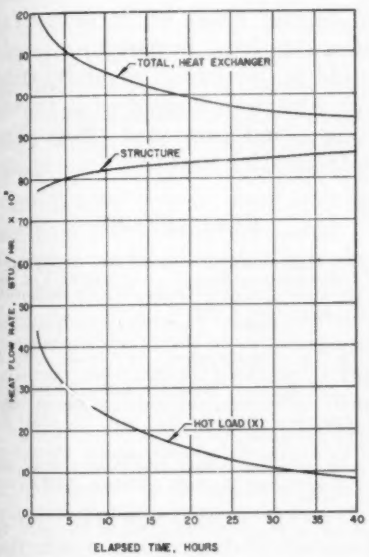


Fig. 10 Heat-flow vs. time with hot load

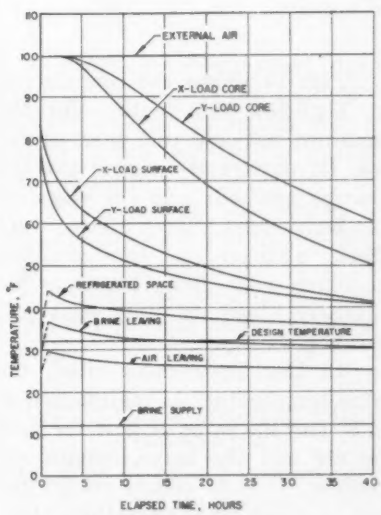


Fig. 11 Dual-hot loads, temperature-time

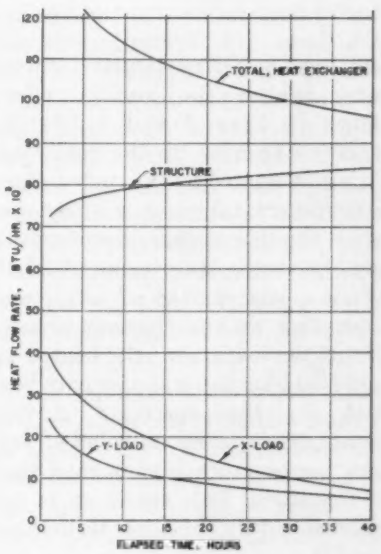


Fig. 12 Dual-hot loads, heat flow

flexibility is possible since in the "pilot" procedure, only the voltage pattern needs to be obtained. Thereby the voltage values at the load slidewires are monitored to govern the current flow which passes through the load surface resistance to the space-voltage bus. As stated before, for the "pilot" system, the electrical value of r_h was set up via r_c of the (r_cC_c) value, in accordance with the required thermal ratio of R'_c/R'_h.

Operation Predictions: Fig. 9 shows the temperature-time history for the case of the hot Load x charged into the empty refrigerated space, of steady-state design value of 32 F, brine-supply at 12 F. The introduction of the hot load causes an immediate rise in the space temperature beyond 32 F, and then tapers off as time passes. The hot-surface temperature, which is the cause of the added heat-load via temperature difference vs the space, likewise falls off with the passage of time, as does the core temperature. As may be expected, brine-outgoing as well as air-leaving temperatures at the heat exchanger also increase, and gradually fall off with the passage of time.

Fig. 10 shows the corresponding rates of heat-flow vs time. Due to the increase in space temperature, the structure heat-flow itself is decreased: the addition of the high rate of flow from the hot load, however, causes the total flow to the exchanger to increase. With the passage of time these effects tend to taper off.

Figs. 11 and 12 show results similar to those of Figs. 9 and 10, but in place of the single hot Load x introduced into the refrigerated space, two hot loads are involved, namely, both Load x and Load y, of *differing* thermal characteristics, with enhanced effects throughout.

Fig. 13 shows the result of the hot Load x brought into the space already partially occupied by the

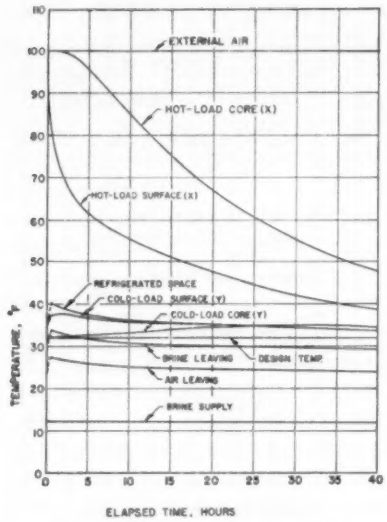


Fig. 13 Hot load brought into cold load space

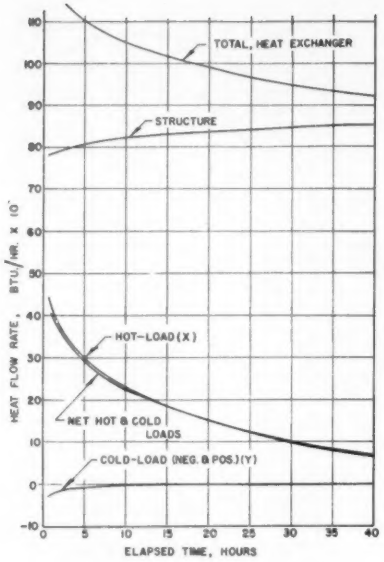


Fig. 14 Heat-Flow for hot-cold load space

cold Load y. Of particular interest here, behind the similar effect shown in Figs. 9 and 11, is the tendency to heat up the cold load. This is indicated by the cold-surface temperature taking a position between the actual space temperature and the steady-state value of 32 F. After a long period of time, and depending on the thermal properties of the loads, etc., the heated-up cold load likewise will dissipate heat back to the space-air therein eventually showing its value of surface temperature higher than that of the space. This condition is not yet reached by the time 40 hr have elapsed.

Fig. 14 shows the corresponding heat-flow rate history for the combined hot and cold load operation. Of particular interest are the *positive* hot-body and the *negative* cold-body heat flows. Their net effect, additive to the modified structure heat flow, produces an increased load to the heat exchanger.

In the three cases cited, the brine temperature is maintained at 12 F, and the rates of flow for both the air and the brine remain unchanged. It is of course recognized that in given plant operations these conditions might well vary. Particularly to be noted is that the brine-supply temperature, often dependent on refrigerating-machine performance, might quite well rise due to the added heat load. Under these circumstances, the space temperature would be further increased, with extended effects on the different heat loads.

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NOMENCLATURE

A = surface area, sq ft
$a = k/(\rho c)$ = thermal diffusivity
c = specific heat, Btu/(lb-F)
c_a = specific heat of air, Btu/(lb-F)
c_b = specific heat of brine, Btu/(lb-F)
C_e = electrical capacitance (farad, microfarad)
C'_s = thermal capacity per unit of frontal area = $c\rho Y$
f_s = Schack heat-transfer function, temperature - difference ratio
E = electrical voltage for analogue
F = difference or resistance ratio
h = heat transfer coefficient, Btu/(hr-sq ft-F)
I = electrical current for analogue
k = thermal conductivity of solid material, Btu/(hr-sq ft-F/ft)
q = heat flow rate, Btu/hr
R' = thermal resistance, referred to 1 sq ft, F/(Btu/(hr-sq ft))
R'_a = thermal resistance of surface air, F/(Btu/(hr-sq ft))
R = thermal resistance, F/(Btu/hr)
R_a = heat exchange resistance of air stream = $1/(W_a c_a)$
R_b = heat exchange resistance of brine stream = $1/(W_b c_b)$
R'_c = thermal resistance of solid path Y in length, = Y/k
R'_h = thermal resistance due to fluid conductance h, per sq ft = $1/h$
r_c = electrical resistance, ohms, proportional to R'_c
r_h = electrical resistance, ohms, proportional to R'_h
t_a = surrounding air temperature F
t_{ex} = external atmospheric temperature, F
t_o = original initial solid temperature F
t_s = surface temperature F
t_x = temperature in solid at any point x, F
Δt = temperature difference, F
W_a = weight flow rate for air, lb/hr
W_b = weight flow rate for brine, lb/hr
U = exchanger overall coefficient of heat transfer, Btu/(hr-sq ft-F)
Y = half-slab thickness, ft
ρ = density, lb/cu ft
τ = elapsed time, hr

Subscripts to denote special applications are used where required.

Noise and vibration

in water piping systems

W. L. ROGERS

Noise implies vibration. Audible noises can exist when vibrating surfaces in contact with the air transfer their vibration displacements to the air, which then conveys the resulting pressure variations to human ears. Whether or not sound is perceived depends upon the frequency and amplitude of the air-pressure fluctuations. The frequency range of hearing is roughly 15 or 20 cycles per second (cps) to 15,000 or 20,000 cps. As for amplitude, Fig. 1 gives some idea of the magnitudes of sound-pressure levels measured in various environments.

Whether a new sound such as might be added by components of a piping system adds anything perceptible to an existing sound level depends on the relative magnitudes of the 2 sounds. A change of the order of one decibel must exist before a change is perceptible. Fig. 2 indicates the total sound-pressure levels in decibels obtained by adding 2 sounds. It is apparent that a new sound may have to have appreciable magnitude if it is to have a noticeable effect. For example, adding 2 sounds of equal levels results in an increase of only 3 db, while adding a 50 db sound to an existing level of 60 will produce no perceptible change at all. Probably it should be mentioned here, too, that average human ears are more sensitive to some frequencies than to others. Frequencies in the range of 1000 to 5000 cps are usually most easily heard, with sensitivity falling off at higher and lower frequencies.[1]

A list of the pertinent vibrating surfaces which produce noise in water piping systems includes (a) the surfaces of pumps and motors, (b) the surfaces of pipes and piping components, and (c) the surfaces of building panels to which pumps, motors, pipes, or piping components are attached.

The following discussion relates to sources, transmission, magnification, and reduction of oscillations which may result in excessive vibrations of these noise-producing surfaces.

SOURCES OF NOISE-PRODUCING VIBRATIONS

Vibrations of piping systems originate at one of two places: (1) within the fluid, or (2) in some solid (structural) part of the system. Examples of fluid sources are the fluid pressure pulsations from pumps, from flow through piping components, and from water hammer. Structural vibration sources include unbalanced rotors in pumps and motors, frictional forces, electrically-induced mechanical forces, and loose parts in the system.

It is well known that the vibration produced in the fluid does not remain entirely in the fluid, and that the vibration produced in the structure does not remain in the structural parts of the system alone. There is interaction, and a pressure oscillation in the fluid can cause excessive lateral pipe vibration under certain circumstances. The interaction is much stronger for water-filled pipes than for gas or vapor-filled pipes, an important factor being the relative compressibilities of the fluid medium and the material of the pipe. The pipe dimensions are of importance, also, for they influence the stiffness presented by the pipe wall to the pressure variations in the fluid.

A self-excited or self-sustained type of oscillation can be encountered in certain types of valves. The essence of this type of oscillation is that the vibrational motion itself includes actions which tend to sustain the vibration. For small amplitudes, energy is fed into the vibrational motion to increase it, while for much larger amplitudes energy is removed so that the vibration is decreased. The resulting eventual steady state of oscillation is sometimes called a limit cycle. For small amplitudes, the damping is negative, while for large amplitudes damping becomes positive.

The oscillations or collapse of gas or vapor bubbles in the liquid of the system can be potent sources of noise. Bubble oscillations are produced as a bubble is carried by the stream through a region of varying pressure. The varying pressure on the bubble constitutes forced oscillation of the bubble, but the natural frequencies of oscillation of the bubble are excited as well. It is possible that flow noises produced from other causes, such as pumps, can be intensified or damped by the presence of bubbles, depending upon the forcing and natural frequencies. The noise of cavitation requires the presence of cavities in the liquid. A common experience is the elimination of hissing noises resulting from elimination of trapped gas.

In experimental work[2,3] it has been found that sudden enlargement of a flow passage into a low-pressure region results in particularly intense noise generation within the flowing water. Flow through orifices or through partially-closed valves are examples.

[1] Exponent numerals refer to References.

W. L. Rogers is Associate Professor of Mechanical Engineering at Northwestern University. This paper was presented at the Symposium on Hydronics at the 65th Annual Meeting of the American Society of Heating and Air Conditioning Engineers, Philadelphia, Pa., January 27, 1959.

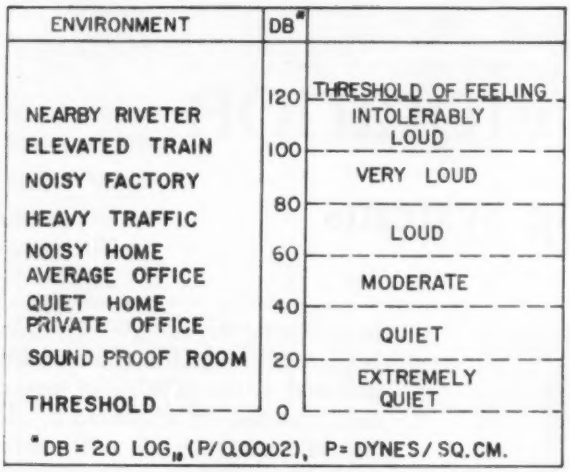

Fig. 1 — Sound levels in representative environments

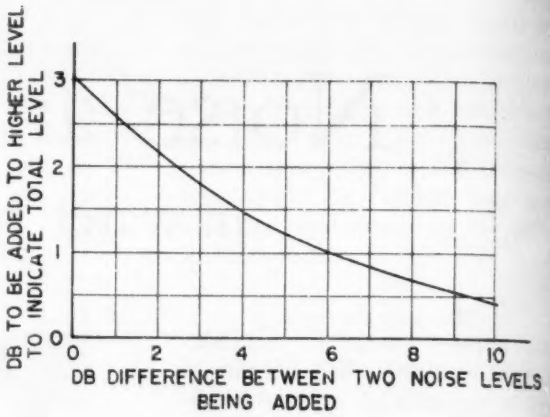

Fig. 2 — Adding two sound levels

It is difficult to draw definite conclusions about the relative importance of the various types of noise sources. However, some reasonable observations can be made. The rather wide range of frequencies inherent in noises produced by flow through most piping components (such as valves and fittings) means that the energy of such noises is not concentrated in particularly narrow bands. Further, the vibrations which do exist seem often to consist of free oscillations started at random intervals as by shedding of eddies and cavitation phenomena.

It would seem that pump noises would constitute a greater hazard for producing intense noise problems at locations away from the source. They are a source of steady frequency, with greater possibility of resonance downstream. In tests, it has been found that the level of noise in the water produced by flow through representative components (such as elbows or open valves) can be of the same order of magnitude as that produced by a representative pump.

TRANSMISSION OF NOISE FROM THE SOURCE

Noise (vibration) can travel along the liquid path, the structural path, or both. The split of sound energy into various paths as a sound travels away from its source depends upon the acoustic impedance of the various paths available. As is the case in the flow of electricity, the greatest flow follows the path of least resistance.

For a sound source in a fluid, the best acoustic impedance match, and consequently the greatest flow of the acoustical energy, is usually along a continuing fluid path. However, the pipe walls surrounding the fluid are not perfectly rigid, and so some of the vibration of the fluid is transmitted to the pipe walls. This fraction is much greater for liquid-filled pipes than for gas-filled pipes. In addition, for structural resonance conditions which may exist, and which are discussed later, the resistance of the pipe to vibration may be reduced, resulting in large pipe vibrations induced by fluid sources. Also, if there is a sudden change of size of the fluid duct path, there can be reflection backward along the fluid path, thereby reducing the amount transmitted.

For the structural sources, the general thoughts are the same. Most of the vibration travels along the pipe or structure, with branching paths carrying vibrational energies in amounts dependent upon the relative impedances of the paths.

The frequency of vibration is an important factor in transmission of noise. As for any vibration, if a natural frequency of a branching path coincides with the frequency of the vibration, the amplitude of vibration at the particular location will be increased. In recent experimental work conducted at Northwestern University under a cooperative arrangement with the ASHAE, but as yet unpublished, it has been demonstrated conclusively that large pipe vibrations can be produced by pulsing the water at frequencies corresponding to the natural frequencies of the pipe and its supporting arrangement.

Dissipation is inherent in any vibration. Friction and hysteresis effects reduce the amplitudes of vibrations as they are transmitted. Thus, dissipation reduces the magnitude of transmitted noise. The inherent dissipation along the fluid path and along the structural path can be quite small in actual installations, so that damping by inherent dissipation cannot be relied upon to be a significant factor in reduction of transmitted vibrations.

The effectiveness of rubber hose in reducing transmitted vibrations varies with its stiffness and its length, as well as with the frequencies present. Soft hose is better than stiff hose, and high frequencies are more effectively damped than low frequencies. Reliable and complete quantitative information is not available, but the length of commercially-available hose required for worthwhile sound reduction is practical in many instances, amounting to a matter of no more than a few feet. Both fluid-borne and structural-borne vibrations can be reduced, usually by different amounts. This is particularly advantageous because of the interaction already discussed. However, there are problems of cost, maintenance, and safety.

Hose reduces fluid-borne sound probably by a combination of pressure relief and internal damping (dissipation) in the rubber as it vibrates with the pulsing fluid. The latter action is quite evident as one touches the surface of soft rubber

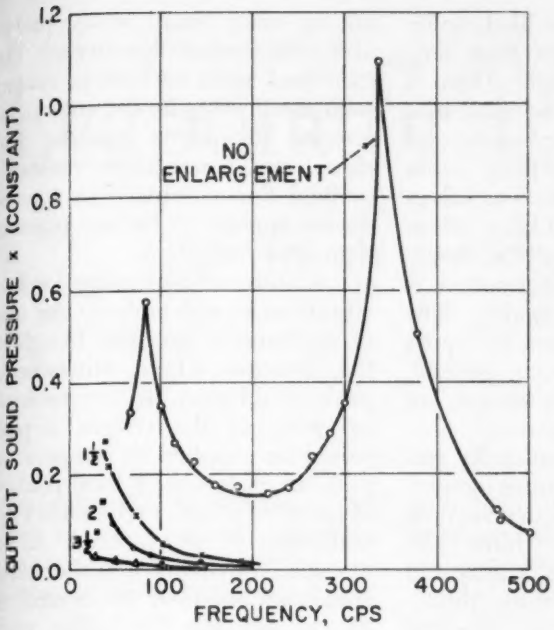

Fig. 3 – Sudden enlargements in 1 in. pipe as low-pass acoustic filters. All lengths 18 in.

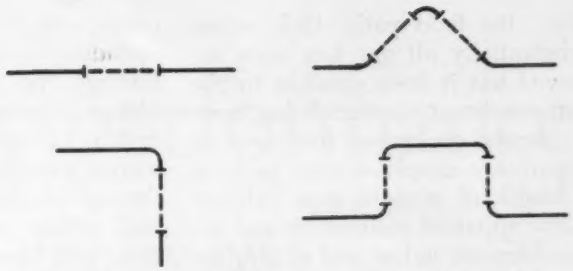

Fig. 4 – Example arrangements of hose inserts to reduce vibration transmission

hose or listens with a stethoscope to the rapidly-diminishing sound as one moves downstream from the entrance to the tube.

Hose impedes transmission of structural vibration by introducing flexibility in the structural path.

A sudden enlargement in a pipe line acts as a low-pass, high-frequency cut-off filter for fluid-borne vibrations. Fig. 3 taken from an earlier paper[3] illustrates this filter action. Filter sizes are practical. Only a limited amount of design information is available at present for such water filters, some of it obtained at Northwestern University, but as yet unpublished. The greater the enlargement, the lower the frequency above which frequencies are cut off.

In general, to stop transmission of noise in liquid piping systems, it is necessary to insert a block in the path of transmission. The block can reflect the vibration, damp it by dissipation, or counteract it. This last technique will not be pursued here. It has its counterpart in the well-known dynamic vibration absorber discussed in many books on vibrations. The method of application of the idea to piping systems needs further study.

RESONANCE

A piping system inherently contains a multitude of opportunities for resonance with source disturbances. In the fluid path are organ-pipe effects, and in the structural path are vibrations of pipes, components, and the masses to which they are attached.

A vibrating surface which is radiating sound produces more sound as it vibrates at greater amplitude. It has been amply demonstrated that greatly increased lateral vibration of a piping span can be produced if the lateral natural frequency of the span coincides with the frequency of either fluid or structural vibration sources. This in itself might not produce a comparably greater sound, particularly for small pipes. It is to be noted that it is not the vibration of the air caused directly by vibration of the string of a piano which is responsible for the loud sound. It is the vibration of the much larger surface of the sounding board which is primarily responsible. It is true that the vibration of the string causes the sounding board to vibrate, and that larger string vibrations cause larger sounding-board vibrations. The analogy to pipes (as strings) connected to building panels (as sounding boards) is obvious. The type of pipe supports and the characteristics of the panels obviously affect the elasticity of the system and hence the natural frequencies.

Although the work has not yet been published, in the course of studies at Northwestern University the natural lateral frequencies of both empty and water-filled piping spans have been calculated and spot-checked experimentally. Several lateral modes in several pipe sizes have been considered. Supports at the ends were considered fixed-fixed or free-free. A notably interesting result is that a whole host of natural frequencies which would represent actual practice (such as to span and probable support stiffness) lie squarely in the audible-frequency range. It seems probable that this result is of considerable importance in attempting to understand the noise problems associated with piping systems.

Presumably, the fluid organ-pipe resonance effects in a system can cause magnification of the amplitude of the pressure variations of fluid sources such as pumps. Thus, excitation could be increased for vibration of pipes and their attached panels, and a noise problem might become significant which might have been negligible without the fluid resonance effect. Simple organ-pipe resonant frequencies for liquids (and gases) can be calculated, and have been checked reasonably well in experiments with water in pipes open at one end and closed at the other. Investigations pertinent to a real system with expansion tanks and closed piping loops still need to be carried out.

Experimental work at Northwestern University has indicated the extreme importance of the presence of gas bubbles in the water when studying pressure-wave travel

along the fluid path. Only when substantially all gas has been removed has it been possible to obtain consistent results. It has been a common occurrence that even an apparently simple system such as a length of straight pipe exhibits many apparent resonances and inconsistencies unless and until quite special care has been devoted to elimination of gas in the system. The probable reason for the observed effects is the presence of a considerable range of bubble sizes. Bubble size affects bubble natural frequency. For given forcing disturbances, some bubble sizes apparently can cause amplification of the fluid vibration, while other sizes can reduce the fluid vibration by a dynamic absorber action.

A somewhat rare occurrence can be visualized in which a fluid source such as a pump comes into organ-pipe resonance within the fluid system, this frequency coinciding with that of some pipe span and supporting panel. A loud noise would be expected. More usual problems probably do not include all these simultaneous happenings.

It might be worth noting that the presence of a range of frequencies produced by flow noises offers the possibility of selective magnification of some of them in particular systems. However, it seems reasonable to suppose that pumps usually will present the greater hazard, because of their concentration of sound at particular frequencies.

SOME MEANS OF MINIMIZING NOISE

General considerations must include reducing the source amplitudes; avoiding amplification by resonance; and minimizing transmission by means of filters or reflectors, damping by dissipation, or counteracting devices.

Low-noise pumps must have properly-designed flow passages, must be balanced, and must be operated by balanced electric motors. In addition, soft isolation mountings are commonly used to block structural transmission. A low-noise system also requires use of reasonable flow velocities. No absolute rule can be given here. In general, tests have indicated[2] that flow through the components (such as elbows and valves) of a piping system are more likely to be offending noise-makers than flow through the pipe itself. Thus, a higher velocity could be used in a system of fewer components and section changes. Throttling action through obstructions such as valves and orifices is particularly offensive, and especially on the downstream side. Such information is important when designing flow passages in pumps, too, for quiet operation. It appears, in general, that gradual pressure drops are preferable to sudden ones.

Avoiding amplification by resonance obviously requires consideration of forcing frequencies and natural frequencies. Motor and pump speeds and the number of pump impellers determine significant pump frequencies. Type of piping component and velocity through it determine frequencies of this flow source of noise.[2] The presence of bubbles also is of importance. The pertinent natural frequencies might be controlled by altering motor, pump, and pipe mounting stiffnesses; lengths of piping spans; lengths affecting organ-pipe resonances; by elimination of random bubble sizes; and probably by adjustment of the vibrating masses, where feasible.

Transmission can be reduced by using proper rubber-hose inserts, which can block both fluid- and solid-borne vibration when properly arranged. Fig. 4 shows several arrangements. While offering increased flexibility and therefore seemingly greater damping action, hose inserts also introduce hazards of new natural frequencies, and, when curved, they can impart structural vibration to the attached pipe when fluid pressure fluctuates at the bend. Hoses also have obvious limitations.

Sudden enlargements act as low-pass filters, stopping transmission along the liquid path of frequencies above a value which is affected by the size of the enlargement. Fig. 3 illustrates the effect, but more design detail for water-filled systems is needed. Other types of filters are possible, too, including high-pass and band-pass. These have had only cursory attention as related to water piping systems.

Damping devices can be constructed which divide the flow among many small tubes, preferably with laminar flow in each. The increased solid surface in contact with the flowing fluid results in increased dissipation due to fluid shear, and hence more resistance to fluid fluctuations. This type of device appears to be less practical than some others.

Counteracting devices for fluid vibrations would embody the idea of applying a pressure trough to the location where otherwise a peak would exist. In the mechanical parts of the system, a push would be supplied to counteract a pull. In each case, a local position of essentially zero variation (a local node) would be enforced. As an example, a dynamic absorber effect could be obtained by control of gas bubble size, or, more practically, by the use of equivalent elastic elements in the system.

CONCLUSION

It has been the intent here to summarize in a qualitative manner some of the observations of the author concerning the problems of noise in water piping systems. The bibliography and reference titles listed at the end of the paper contain additional references and a considerable amount of pertinent quantitative information which it was felt need not be repeated here, as it is readily accessible.

The author is associated with a continuing cooperative relationship between the ASHRAE and Northwestern University which has as its purpose the study of noise in liquid piping systems. Future reports arising out of this study will present information concerning many aspects of the problem which have only been mentioned in this brief survey.

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Evaporator frost performance

in an ice cream display case

PAUL P. OMELIANCHUK
Member ASHRAE

With automatic time-initiated defrost devices, cabinet engineers face a problem in selecting the number of defrost cycles per unit time. Open top (reach in) low temperature display cases, where the means of cooling is more or less exposed to existing seasonal humidity, have but made the problem more complex. There is an added variable of geographic location.

Thus, in a given geographic location, the open top display case described herein has a seasonal difference in the number of required defrost cycles of approximately once in 480 hrs to once in 6 hrs or once in 20 days to once in ¼ day. This extreme difference leads to a compromise setting of perhaps two defrosts per day resulting in a higher rate of useless defrosts at low humidities with still an inadequate number at high humidities.

It was with this problem in mind that an effort was made to improve frost performance of a fin and tube evaporator in a gravity type display case and thereby improve the annual defrost cycle efficiency.

The difficulty is admittedly less important in an air conditioned store. However, there are now and probably will be for many years to come marketing areas where humidity problems in low temperature display cases will not be solved completely by air conditioning. Such areas may include geographical locations where the air dry bulb temperature is sufficiently low over an extended period of time to preclude air conditioning, but the wet bulb high enough to invite frost problems during refrigeration at low temperatures.

In addition, it is anticipated that in the near future a reliable defrost control will be developed to initiate the defrost cycle on the basis of humidity and not of time. This can be accomplished by starting defrost from a predetermined signal furnished by the frost on the coil. With this type of device available it would be desirable to obtain maximum coil output for the greatest possible interval at high humidities.

Our investigation on the effect of fin arrangement of a gravity type cooling coil in an open top ice cream display case, related to the necessary number of defrost cycles required when the cabinet is operated under high humidity conditions. The scope of this investigation is limited—no attempt was made to study all of the factors affecting optimum design.

This discussion deals only with a cooling coil in one commercially available cabinet so that such factors as coil size and shape and air flow arrangement were fixed. The results of this investigation could be changed by a change of any of several design factors. It is important to note that ice cream was

Fig. 1 Gravity type, self contained display case used in frost performance evaluation tests

Paul P. Omelianchuk is Assistant Chief Engineer, Schaefer Inc. This paper was presented at the Commercial Refrigeration Conference as "Frost Performance Evaluation of a Fin and Tube Evaporator in a Gravity Type Ice Cream Display Case" at the 45th Semiannual Meeting of ASRE in New Orleans, December 1-3, 1958.

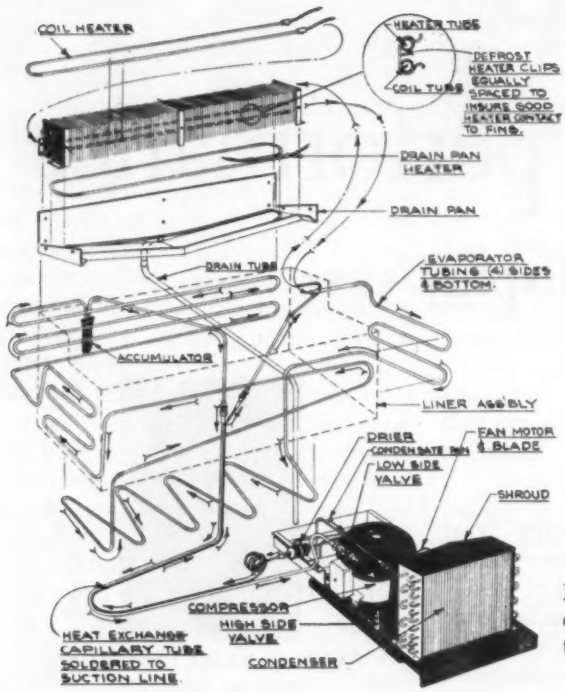


Fig. 2 Skeleton view of refrigerating system in display case

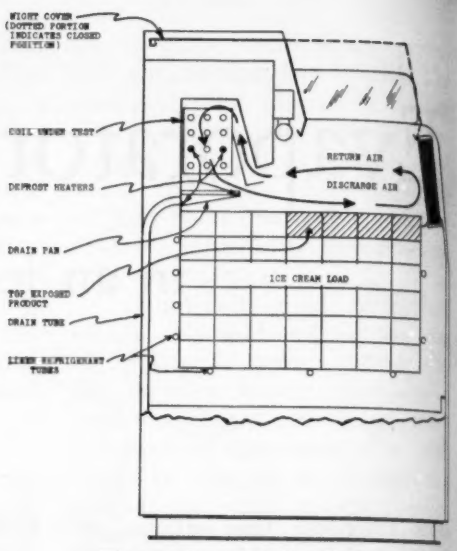


Fig. 3 Cross-sectional view of test cabinet showing convectional air path, defrost heaters and product load

used as the load in this evaluation. Since this product is one of the most difficult to maintain, from a temperature as well as a quality standpoint, the product had influence in the conclusions.

Description of Display Case—The data in this report were obtained from a semi-automatic defrost cabinet 6 ft long equipped with a ¾ hp self-contained capillary tube refrigerating system. The front of the cabinet exterior is shown in Fig. 1. A skeleton view of the refrigerating system is shown in Fig. 2. During the refrigerating cycle the refrigerant is metered by capillary tube to the fin coil, then through the lower wrap around tubing and accumulator, then back to the compressor via the combination suction line-heat exchanger.

A cross-sectional view showing the gravity air circulation through the fin coil is given by Fig. 3. The warm air rises in the return air duct, falls by gravity through the fin coil where it is dehydrated then cooled, and continues through the discharge opening towards the front of the cabinet then back to the return duct completing a cycle.

The defrost cycle is time initiated with temperature termination. Defrosting is accomplished by stopping the compressor and energizing the coil and drain pan heaters. Thermostat termination provided the means of comparing defrost cycle efficiency since the amount of frost collected on a given coil could reflect itself in defrost time with equivalent wattage input.

The fin coil has a two-fold re-

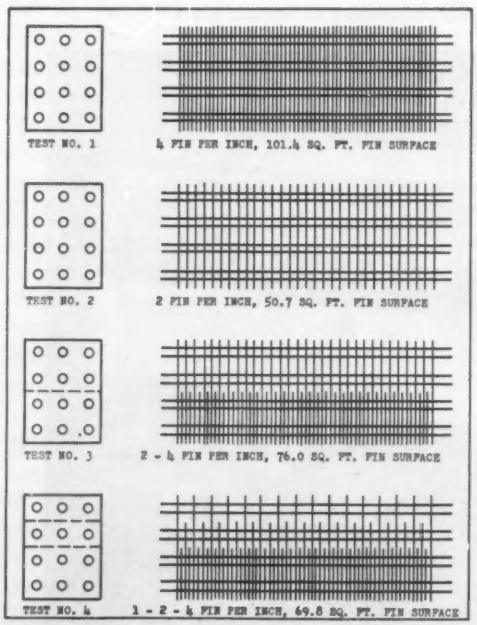
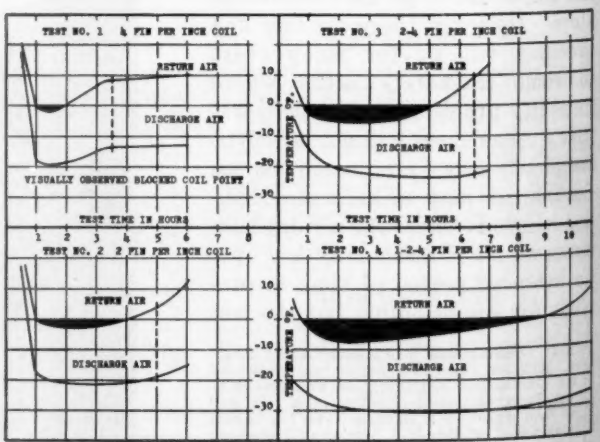

TEST NO. 1
4 FIN PER INCH, 101.4 SQ. FT. FIN SURFACE
TEST NO. 2
2 FIN PER INCH, 50.7 SQ. FT. FIN SURFACE
TEST NO. 3
2 - 4 FIN PER INCH, 76.0 SQ. FT. FIN SURFACE
TEST NO. 4
1 - 2 - 4 FIN PER INCH, 69.8 SQ. FT. FIN SURFACE

Fig. 4 Description of fin arrangement for each test

Fig. 5 Return and discharge air vs. time for each test. Shaded area is that portion of return air temperature below 0 F and represents the useful refrigerating effect accomplished between defrost cycles


TEST NO. 1 4 FIN PER INCH COIL
RETURN AIR
DISCHARGE AIR
VISUALLY OBSERVED BLOCKED COIL POINT
TEST TIME IN HOURS
TEST NO. 3 2-4 FIN PER INCH COIL
RETURN AIR
DISCHARGE AIR
TEST TIME IN HOURS
TEST NO. 2 2 FIN PER INCH COIL
RETURN AIR
DISCHARGE AIR
TEST NO. 4 1-2-4 FIN PER INCH COIL
RETURN AIR
DISCHARGE AIR

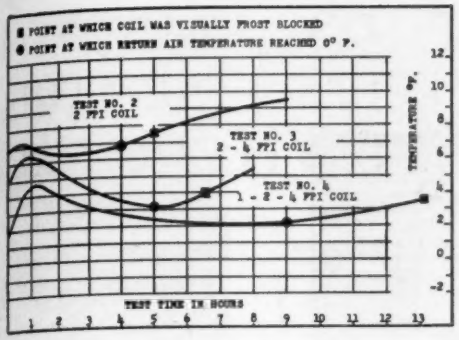

● POINT AT WHICH COIL WAS VISUALLY FROST BLOCKED
● POINT AT WHICH RETURN AIR TEMPERATURE REACHED 0° F.
TEST NO. 2
2 FPI COIL
TEST NO. 3
2 - 4 FPI COIL
TEST NO. 4
1 - 2 - 4 FPI COIL
TEMPERATURE °F.
TEST TIME IN HOURS

Fig. 6 Average top layer exposed ice cream temperature vs. time for tests 2, 3 and 4

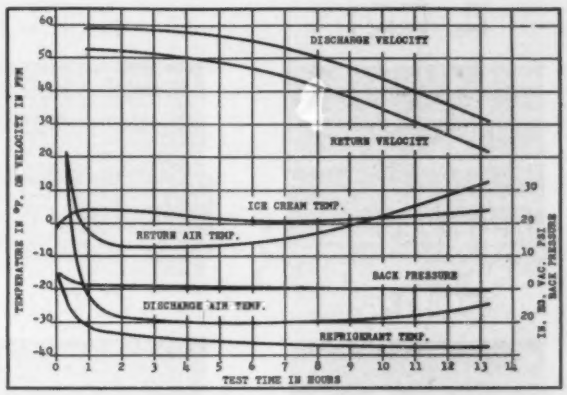


Fig. 7 Plotted results of test 4

sponsibility: (1) To pick up the water vapor which migrates into the cabinet as a result of convectional and diffusional movements and (2) to cool the air circulating over the coil to a temperature low enough to properly refrigerate the top of the ice cream load. The lower coil or wrap around tubing on the liner for the most part offsets the heat gain through the sidewalls and bottom, thereby keeping the upper coil capacity at work on the upper portion of the load.

Description of Tests—The cabinet was loaded with pint packages of ice cream to the load level shown in Fig. 3. Tests were performed in a humidity room under constant conditions of 80 F dry bulb and 70 % relative humidity. Room air velocities in all of the tests were somewhat high and contributed towards an accelerated test.

After a 16 hr stabilization period with the night cover closed, the coil was defrosted, the night cover opened and readings taken every half hour. These included product* temperatures, discharge and return temperatures, room dry and wet bulb temperatures, discharge and return air velocities and a photograph of the coil showing the frost build up progress. Each test was concluded when, by inspection, the coil appeared 100% blocked. Four such tests were run, each with a coil with different fin arrangements. (See Fig. 4.) Test No. 1 had a straight 4 fin per in. coil; test No. 2, a straight 2 fin per in.; test No. 3, a staged 2-4 fin per in.; and test No. 4, a staged 1-2-4 fin per in. The cabinet position in the test room remained unchanged for all 4 tests thus insuring the same test conditions for each coil.

* It is also important to note in Fig. 3 that the shaded product area represents the top exposed product from which temperature measurements were made in the tests.

Discussion of Results—The above tests were run in the order shown and scheduled only after thorough evaluation of the previous test. Fig. 5 shows the plotted results of discharge and return air temperatures vs time. The shaded area between the zero degree datum line and that portion of the return air curve lying below has been selected as a comparison point and is referred to in this discussion as the

TABLE I—TABULATED DATA AND CALCULATIONS

	Test No. 1 4 FPI	Test No. 2 2 FPI	Test No. 3 2-4 FPI	Test No. 4 1-2-4 FPI
1. Secondary (fin) surface in sq ft	101.4	50.7	76.0	69.8
2. % Surface based on 4 FPI	100	50	75	68.8
3. Time for return air to warm up to 0 F (hr)	2.0	4.0	5.0	9.0
4. Time for coil to frost block (hr observed time)	3.5	5.0	6.5	13.2
5. Return air F at observed blocked coil time	8.5	10.0	12.0	12.0
6. Lowest average return air temperature obtained F	—2	—3	—5.5	—7.0
*7. Average blocked coil product temperature F	NR	6.8	4.0	3.8
*8. Lowest average product temperature	NR	6.2	3.2	2.1
*9. Average product temperature @ 0 F return air temperature	NR	6.3	3.5	2.3
10. Net time (hr) return air is maintained below zero (Fig. 5)	1	3	4	8
11. Ratio of time (10) above to defrost plus recovery time of 1½ hr	.666	2	2.67	5.34
12. Improvement ratio based on Test 1. (11) ÷ .666	1	3	4	7.5
13. Pounds of frost collected per defrost cycle	.480	1.380	1.68	2.75
14. Time in hr for complete defrost	.45	.45	.5	.534
15. Defrost cycle efficiency	8.6	24.6	27.0	44.0
16. Theoretical no. of defrosts per day required based on (3) above	12	6	4.8	2.67
17. Sensible heat gain per day based on 14, 15 & 16	12,300	5,090	4,370	2,720
18. % of daily condensing unit capacity	28.4	11.8	10.1	6.3

* Top exposed product

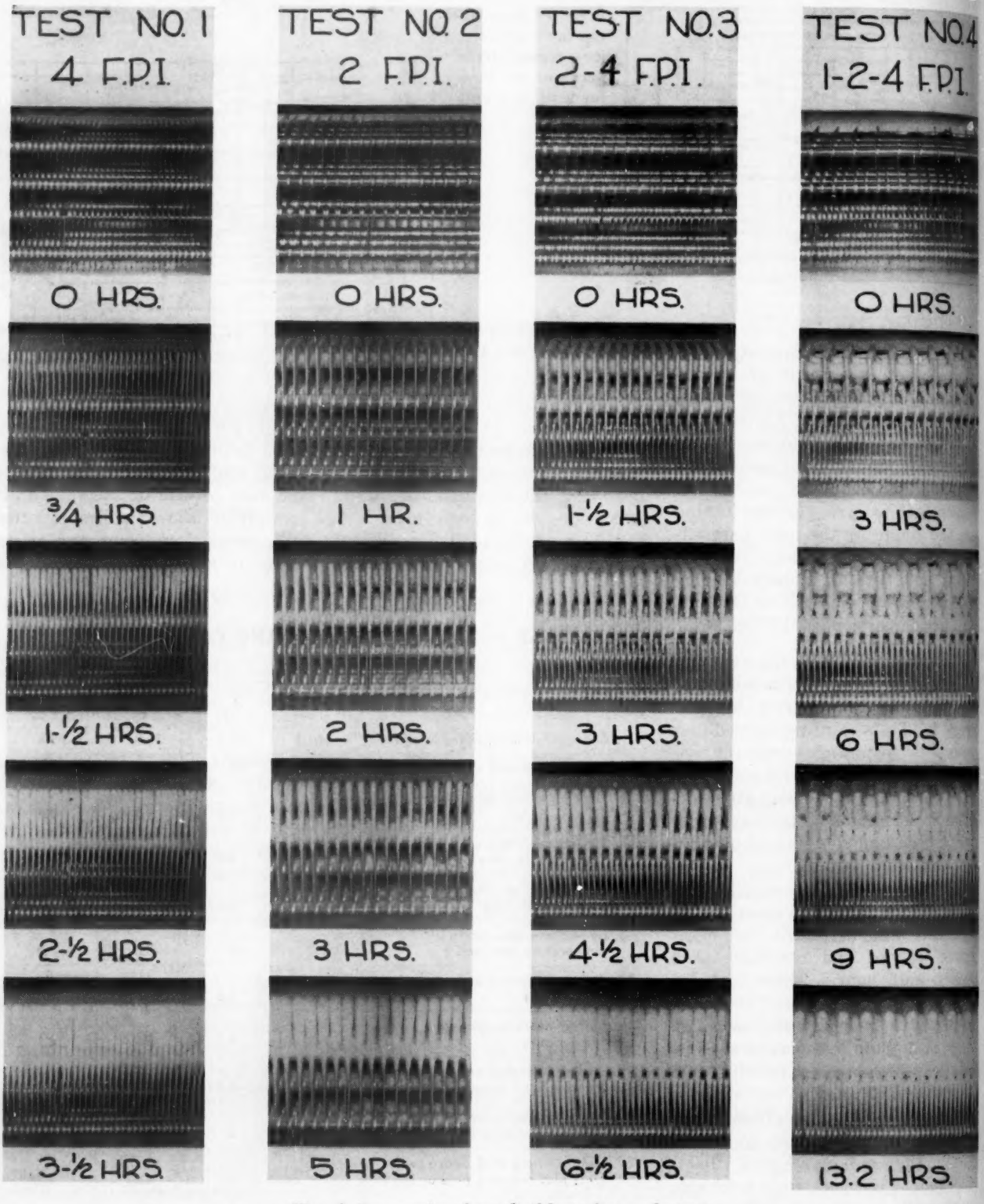


Fig. 8 Progressive frost build-up for each test

useful refrigerating effect produced by the upper coil between defrost cycles.

Fig. 6 shows a comparative plot of average ice cream temperatures vs time for Tests 2, 3 and 4. Ice cream temperatures for Test 1 were not recorded. Fig. 7 shows a complete plot of the recorded data of Test 4. This was the only test in which discharge and return velocities were taken. These curves represent a fair average of points and indicate the loss in air flow as a function of frost build up. From the above plotted data improved performance is evident.

Staged fin spacing, which provides the entering air side of the coil with maximum fin spaces, with progressively smaller spaces for the stages thereafter, first dehydrates the entering air, then cools it in the latter stages to its ultimate temperature and thereby increasing the amount of frost the

(Continued on page 122)

Research program to be expanded greatly

BURGESS H. JENNINGS

Director of Research
ASHRAE Research Laboratory

Members of ASHRAE can truly look for greater return from research activities as, with the merger completed, there is now a broader base on which to develop and grow. An initial step in this connection has already been taken with the expenditure rate for research being raised to a higher figure and the planned budget for the next fiscal year, starting July 1, being set at $325,000. This is a definite increase over the combined research budget of the two Societies which, before the merger, was approximately $263,000. The increase represents only an initial step, as the research program is being oriented to expand to the half-million dollar level within the space of a few years.

The research program of ASHRAE is under the direction of the Research and Technical Committee. This consists of twelve members, with four members to be elected every year, to serve for a term of three years. At this time, the initial membership of the Committee can be announced with the present appointments being made on an interim basis to run until the annual meeting in June, 1959, following which the Board of Directors will appoint members for the Committee on the three-year pattern. The complete Committee consists of:

R. C. Jordan, Chairman	N. B. Hutcheon
E. P. Palmatier, Vice Chairman	W. T. Pentzer
D. D'Eustachio	E. F. Snyder, Jr.
H. C. Diehl	R. M. Stern
S. F. Gilman	W. F. Stoecker
F. K. Hick, M.D.	E. F. Wolfert

Because of the many problems resulting from the merger and the consequent expansion of the research program, President Cecil Boling is also inviting a number of consultants to serve in an advisory capacity to the Research and Technical Committee, and the following consultants to the Committee have been invited to serve:

P. R. Achenbach	A. M. G. Moody
A. P. Boehmer	N. Nelles
D. L. Fiske	H. B. Nottage
W. C. L. Hemeon	F. G. Peck
A. Giannini	E. N. Robertson
E. N. Johnson	P. H. Yeomans

In the past, ASHAE has used Technical Advisory Committees to work with the Committee on Research to give specific help to the Committee in fields of research interest. These committees have been of inestimable help to the research program, and to continue the fine type of work that they have done, similarly oriented Research Advisory Committees will be appointed for ASHRAE. These, of course, are special-function committees of the Society. It is expected that the Chairman of the Research and Technical Committee will make the Research Technical Committee appointments in the immediate future. One of these committees will be an Interim Research Advisory Committee on Refrigeration which will act as an overall guiding committee for the inauguration of projects directly related to refrigeration. Ultimately, this will be replaced by the specific committees which will be formed to give coverage to research activities of this field.

The officers and directors of ASHRAE have indicated that the present research programs of the Society will be actively pursued with other programs being added as rapidly as feasible. Consequently, some of the forward planning made by committees at the final meeting of ASHAE should be of interest here. Space does not permit detailed comments for all committees, so only some selected items will be listed.

The Technical Advisory Committee on Sound and Vibration Control indicated the immediate need of determining performance parameters for acoustical lining in ducts. Associated with the activities of this Committee, but yet independent of it, was the Joint ASRE-ASHAE Committee on Equipment Sound Testing. At this time, the work of the Committee has progressed to the point where a third draft of a Standard for Measuring the Sound Radiated from Unitary Equipment by the Reverberant-Room Method is now ready for circulation to members and to industry for criticism. This Committee also is exploring the need of preparing a standard for measuring the acoustic power delivered by a fan or air-moving device to its connecting duct system. It is expected that this Committee

(*Continued on page* 96)

Surface odor adsorption

and retention properties measured

W. F. HOPPER

Results of a study to find a suitable method of measuring odor adsorption and retention characteristics of surfaces are discussed herein. The method adopted is an application of the syringe technique. While data are preliminary with regard to characteristics of the tested materials (cotton, wool, nylon, rayon and wood strips), the method used appears to constitute a satisfactory procedure. Test data suggest that maximum odor adsorption rate of these fabrics during a 24-hr period in odorous air occurs at about 75 F and 50 percent relative humidity.

In order to develop a suitable method of measuring odor adsorption and retention characteristics of various surfaces several fabrics were tested. Cotton, wool, nylon and rayon were chosen as the test fabrics, and iso-amyl acetate was used as the odor agent. Data on the odor retention properties of the first three fabrics are included here; however, data on rayon are not presented because the rayon samples did not adsorb enough odor to be measured.

The method developed and explained is essentially an application of the syringe technique[1] of odor evaluation and retention on various surfaces which have been exposed to odorous atmospheres. All steps in the procedure were carried out under controlled environmental conditions.

TEST APPARATUS

Test Rooms. Preparation of samples and odor appraisals were made in two test rooms[2], separated by a corridor, 8 x 12 x 8 ft high and lined with porcelain-enameled steel having low odor-adsorption qualities. Test room doors were weatherstripped to reduce infiltration of outside air. Inlet air was filtered and deodorized by passing through one bank of mechanical filters, an electrostatic precipitator, a second bank of filters, two banks of activated carbon cells, and finally, through a high-resistance, high-efficiency filter. Air from the rooms was not recirculated but exhausted to outdoors. Three separate air conditioning systems, each with the necessary equipment for cooling, heating, and humidification, served the rooms and the corridor. Air flow was regulated by adjustable dampers on the inlet side of the supply fans and metering orifices were used to measure flow rates.

Odor Adsorption Apparatus: Fabric samples were prepared for test by hanging in a glass jar, 11 in. in diam and 2 ft high, which was fitted with a tight cover. An open dish of odorant was placed in the bottom of the jar, and a second dish of desiccant solution was provided to maintain the relative humidity at the desired level. A small fan circulated the air within the jar.

Bottles. Throughout the program, glass bottles ½-gal in size were used for storing and testing the odorized fabric samples. The bottles were large enough to permit several openings before enough dilution took place to alter test results. Odor transfer to or from the corks was eliminated by wrapping the corks in frequently replaced aluminum foil.

Bottle Cleaning Apparatus. Noxious bottle odors were a major problem which appeared during the early stages of the test program. These odors had to be eliminated because they tended to mask or counteract the odor which had been intentionally introduced into the bottles. To reduce the bottle odors to a harmless level, the bottles were washed with dairy brushes, using an effective odorless detergent, and were boiled for 5 min. After boiling, the bottles were placed neck down on a rack, and air which had been filtered and passed through activated carbon, was blown into them.

Syringes and Accessories. For the syringe method of odor evaluation, twelve 100-ml syringes, twelve syringe caps, one 2-ml syringe, one

[1] Exponent numerals refer to References.

W. F. Hopper is a Research Engineer, ASHRAE Research Laboratory. This paper, prepared for presentation at an early ASHRAE meeting is the result of research carried out at the ASHRAE Cleveland Research Laboratory.

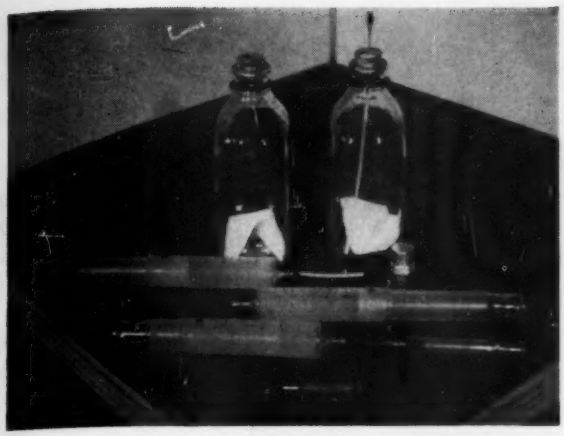


Fig. 1 Equipment used in syringe method of odor evaluation

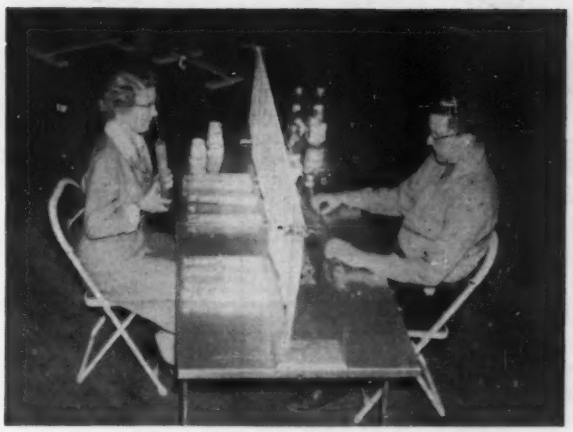


Fig. 2 Odor evaluation test bottles and syringes

10-ml syringe and one transfer needle were used in conjunction with an individual nozzle for each panel member. The transfer needle consisted of a needle with a mating head on each end to permit the joining of two syringes. The extra mating head was silver soldered in place.

The individual nozzle consisted of a Luer Lok fitting for an 18-gage needle from which the needle had been removed and to which a plastic tube had been attached. The syringes and accessories are shown in Fig. 1.

Washing of the syringes followed the same procedure that was used on the bottles.

TEST FABRICS DESCRIBED

The fabrics chosen for test were matched as closely as possible. The factors considered were weave, thread count, finish and weight per square yard. Since the fabrics were not custom made, but were obtained through standard sources, there were minor differences in all the factors with the exception of the weave. Table I lists the properties of the fabrics used in this study.

Odorants. Hexanoic acid was the first odorant used in the testing. This odorant, which may be best described as "goaty," was not only unpleasant to smell, but reacted with the metallic parts in the odorizing jar. For these reasons, its use was discontinued.

To replace hexanoic acid, iso-amyl acetate was chosen because it had a strong and not unpleasant banana-like odor, and was not corrosive. Since the odor of iso-amyl acetate is extremely perceptible, it was necessary to dilute the odorant so that better control of the samples in the loading tank could be obtained. Benzyl benzoate, a nearly odorless diluent, was used to reduce the vapor pressure of the odorant to 10 percent of its original level. The mixture used was 13 parts by weight of iso-amyl acetate with 191 parts by weight of benzyl benzoate. With this dilution at 75 F and 50 percent rh, the odor level of the iso-amyl acetate in the loading tank, in terms of threshold units, was reduced to one million.

Threshold Concept. The term threshold is used as a measure of the concentration of an odorous mixture. The threshold level of odor perception is the lowest concentration of odorous air mixtures which, when mixed with odor-free air, can be consistently identified. If an odorous mixture such as the one in the loading tank has to be diluted to 1 ppm to reach a threshold level, then the original odorous air is one million times more concentrated than the threshold-level mixture or it has an odor level of one million thresholds.

Syringes loaded with odor-laden air were prepared for classification as to odor level by a dilution process using odor-free air. By varying the amounts of odor-free air added to the odor sample under investigation, it was possible to adjust the odor concentration in the syringes until the odor was just recognizable or until the recognition threshold level was reached. For example, with several syringes loaded from the same odor source, one diluted with nine parts of air to one part of odor-laden air would impress on an observer a lower sensation of odor level than would a similar syringe loaded with three parts of odor-free air and one part of odor-laden air.

Consider the contents of the 100-ml reference syringe used in the investigation. At the nine dilution loading, it would contain 10 ml of odor-laden air and 90 ml of odor-free air, while at a dilution of three it would contain 25 ml of odor-laden air and 75 ml of odor-free air. If the odorant in the syringe with the nine dilution loading was just perceptible, the undiluted odorant would have a threshold level of ten, nine-dilution plus one. If, however, the original odor were

TABLE I—PROPERTIES OF FABRICS TESTED

Fiber	Finish	Weight—Oz/Sq Yd	Weave	Thread Count
Cotton	Greige	7.30	Plain	48 x 48
Wool	Scoured & Fulled	7.5 to 8.1	Plain	35 x 41
Nylon	Greige	8.0	Plain	38 x 38
Rayon	Greige	7.60	Plain	38 x 38

TEST: 12 N DATE: MAY 22 1958 MATERIAL: NYLON
ODORANT: ISO AMYL ACETATE
LOADING TIME: 24 HOURS LOADING CONDITIONS D.B. 75 F R.H. 50%
AIRING TIME: 240 MINUTES AIRING CONDITIONS D.B. 75 F R.H. 50%
EVALUATION CONDITIONS D.B. 75 F R.H. 50%

PERCEIVE / NAME	YES	NO	YES	NO	YES	NO	YES	NO	THRESHOLDS	SCORE THIS SHEET
EMC		✓		✓	✓			✓	33	AVERAGE 50
LFS		✓	✓		✓			✓	50	
WFK		✓		✓	✓			✓	33	MEDIAN 41
IMJ		✓		✓	✓			✓	33	
MF	✓		✓		✓			✓	100	WEIGHTED GEOMETRIC MEAN 46
WFH		✓	✓		✓			✓	50	
SYRINGE LOADING	99-1		49-1		24-1		BACK-GROUND			

Fig. 3 Sample fabric odor data sheet

weaker and only three parts of air were needed to dilute it to threshold level, it would be described as a four-threshold odor, namely, three dilutions plus one. For a much stronger odor, the ratio might be equivalent to 0.1 ml of odor-laden air and 99.9 ml of odor-free air or a ratio of 1 to 999; and this mixture, if its odor is just detectable, would be described as having a threshold level of 1000 or merely as being at 1000 thresholds.

PREPARATION OF TEST SAMPLES

Cloth samples, used early in the program for the bottle testing method were of various sizes, ranging from 6 x 6 in. to ¾ x ¾ in., each having an area one-half of the next sample. All samples for the syringe method of evaluation were 6 in. square. Cloth samples were odorized for 24 hr in the loading tank and then aired at the desired environmental conditions for various periods of time ranging from 0 to 24 hr. At the end of the airing period, the samples were stored in tightly corked bottles for at least 24 hr prior to making odor evaluations. The 24-hr periods for odor loading and storage before testing were adhered to—to insure equilibrium conditions.

ODOR EVALUATION PROCEDURE

Bottle Testing Method. At the start of the program, bottle testing was suggested as a method which might be used to determine threshold levels. The bottle method consists of setting up different odor levels in a group of bottles by placing odorous fabric samples of different sizes in them. By removing the corks one at a time and sniffing, each subject selected a bottle in which the odor was at his threshold level.

The method showed numerous inconsistencies which could have been caused by a number of reasons, an obvious one being the possibility of cross-currents of air mixing with the bottle odors at the mouth of the bottle during the process of test sniffing. The syringe testing method was so much more reliable that it was decided to discontinue bottle testing completely.

Syringe Testing Technique. To evaluate the odorous mixture in the bottle, a syringe technique adopted by ASTM[1] was used. The bottle cork was replaced by the cork and tube combination shown in the bottle in Fig. 1. The tube had a Luer Lok fitting on the outer end. Odorant mixture was then withdrawn from the bottle to fill a 100-ml syringe. This syringe became the storage syringe, and the odorant mixture it contained was used as needed for testing. Transfer of the odorant mixture from the storage syringe was accomplished by use of a transfer needle, also shown in Fig. 1. This was connected to a 2-ml or 10-ml transfer syringe, the size depending on the quantity needed. The transfer syringe, after loading, was then attached by the transfer needle to other 100-ml syringes which were prepared for testing.

Each test syringe was charged by introducing a predetermined quantity of odorant mixture and then filling to the 100-ml mark with air from the relatively odor-free test room in which the odor evaluations were made. The syringe was then capped and allowed to stand for at least 15 sec to permit thorough diffusion of the odorant. At the end of this period, a syringe could either be used for a test or its contents could be further diluted. When further dilution was desired, part of the contents were pumped out, the syringe was refilled with test-room air, and after 15 sec, was ready for use.

For each panel member, three test syringes, each containing a different concentration of odorant, were prepared for each test sample. These syringes, together with a fourth syringe containing background odor, were arranged in random order for examination by the panel member. Two such sets of syringes could be examined by a panel member at one sitting before olfactory fatigue became apparent. In addition to the two sets of four syringes, an additional syringe containing known background odor was always available to the panel member for reference.

After a panel member had sniffed and reported his reactions to the four syringes in a given set, the observer prepared additional syringes containing concentrations slightly lower than the lowest concentration originally detected. In this way the threshold level for each panel member was determined more closely. A low screen, shown in Fig. 2, between the observer and panel member, prevented the panel member from observing the preparation of the final syringes and thus being influenced as to the relative concentration of odorant in them.

To evaluate the odor level in a syringe, a panel member connected his personal nozzle to the syringe, and inserted the nozzle into one of his nostrils. The mixture in the syringe was then expelled at a uniform rate over a period of 2 to 3 sec while the breath was held. Holding one's breath was extremely important since breathing would further dilute the sample and inaccurate test data could result. Since the odorant was injected directly into the nose, it reached the olfactory bulb without the aid of breathing.

DATA PROCESSING

The data sheet shown in Fig. 3 has been included as a means of illus-

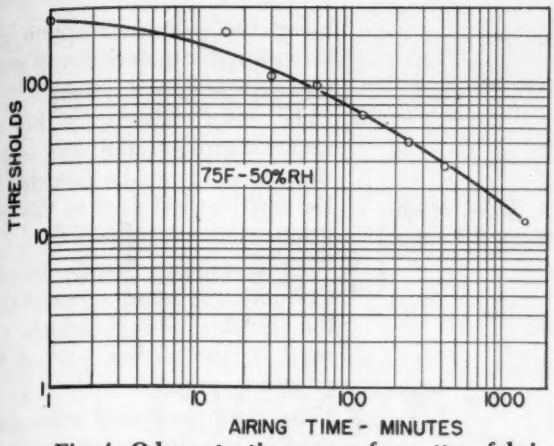


Fig. 4 Odor retention curve for cotton fabric tested

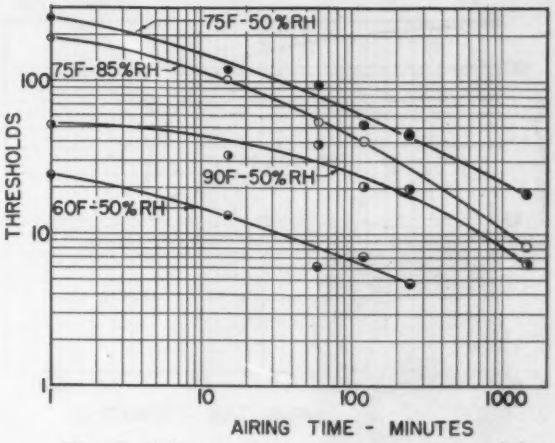


Fig. 5 Odor retention curves for nylon fabric tested

trating the method of gathering data. It may be noted from the figure that subject EMC could detect no odor in the 99-1 and 49-1 dilutions, but did detect the odor in the 24-1 dilution. Thus the dilution ratio which would be at threshold level for this subject was between 24-1 and 49-1. Additional syringes were prepared having dilutions within this range, and a 32-1 dilution was finally judged to be at threshold level, or as noted in the threshold column, EMC's odorant level was 33 thresholds. The dilutions used in the final syringes to pin-point the odor intensity were not noted on the data sheet.

Analyses of the data presented a problem since there were large variations in perception levels among panelists. Averaging the results overemphasized the large variations among panelists, and the resulting average value was open to question. A median method gave less scatter of points but was only moderately successful. A third method, the weighted geometric mean, gave the most consistent record of the test data. The weighted geometric mean (for one airing period) was found by plotting each panelist's perception level at that airing period on logarithmic paper. After the points were plotted, the linear distance from the abscissa to each point was measured in length units, centimeters or inches. The total of these linear distances was then averaged to yield a weighted geometric mean. The plotted points on the following graphs were obtained by this method.

ODOR DESORPTION CHARACTERISTICS

The odor desorption characteristics of the cotton, nylon and wool fabrics tested are shown in Figs. 4, 5, 6 and 7. The temperature and humidity indicated on each curve are those which were maintained throughout the loading, desorption and evaluation processes.

Tests on cotton were made only at environmental conditions of 75 F and 50 percent rh as indicated by the single curve of Fig. 4; other tests were more extensive.

The data for the nylon and wool fabrics are shown in Figs. 5 and 6 respectively. For both of these fabrics, evaluations were made at 60, 75 and 90 F and 50 percent rh, and at 75 F and 85 percent rh.

The odor decay curves for cotton, nylon and wool for 75 F and 50 percent rh have been plotted in Fig. 7 to permit easy comparison. The threshold level at 1-min airing time may be taken as a measure of the odor pickup of each fabric during the 24-hr charging period, while the slope of the curve indicates the rate of odor decay. From Fig. 7 it is evident that both the odor pickup and the rate of odor decay are essentially the same for the nylon and cotton samples tested. The wool sample adsorbed less odor during the 24-

Fig. 6 Odor retention curves for wool fabric tested

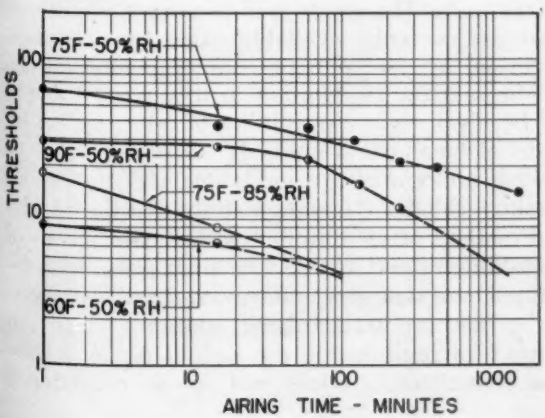
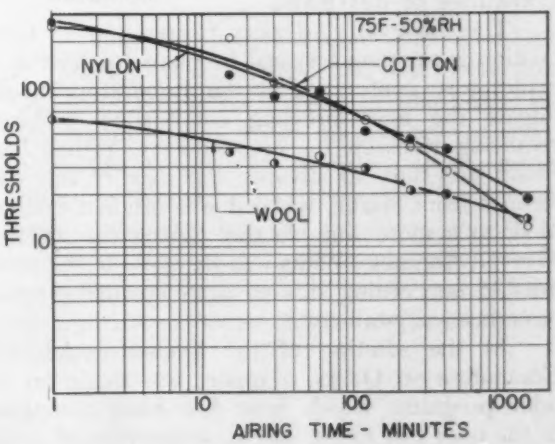


Fig. 7 Odor retention curves for cotton, nylon and wool



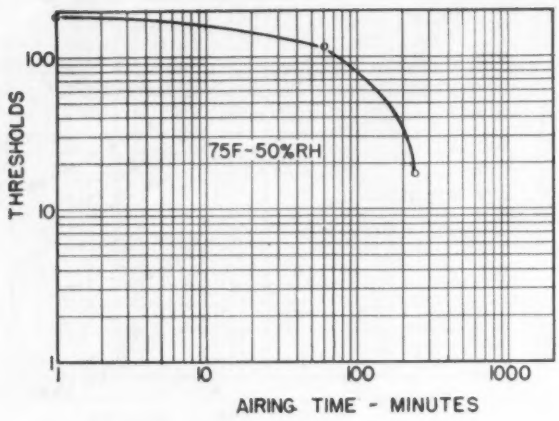


Fig. 8 Odor retention curve for wood

hr charging period but its rate of odor decay was also lower than that of nylon and cotton.

The data presented are for only one bolt of each type of fabric, for a single odorant, and for a limited number of atmospheric conditions. Although no general conclusions can be drawn from the data, some trends are suggested. The curves for both nylon and wool at 60, 75 and 90 F and 50 percent rh (Figs. 5 and 6) show that the maximum odor adsorption during the 24-hr charging period occurred at 75 F, and that the adsorption both at 60 F and 90 F was considerably lower. This implies that maximum odor adsorption occurs on fabrics at about normal indoor temperature.

The curves for both nylon and wool at 50 and 85 percent rh and 75 F show that odor adsorption during the 24-hr charging period was greater at the lower humidity. Additional studies on more types of fabrics with a number of odorants and at other atmospheric conditions are needed to determine whether the trends suggested are typical.

For the purpose of making a rough comparison with a completely different type of surface, three tests were made with wood strips of fir 1½-in. wide and ⅛-in. thick. The total area of the strips comprising each sample was approximately 72 sq. in., the same as the area of the two sides of the fabric samples. The data for these tests are plotted in Fig. 8.

CONCLUSIONS

1. The test method described is a satisfactory procedure for evaluating the odor adsorption and retention characteristics of many surfaces.
2. The odor adsorption by the rayon sample tested was so low that it could not be detected by the test method used in this program.
3. The syringe technique for odor evaluation is more accurate than the bottle method which was used in the earlier part of the program.
4. The data presented were gathered in the process of determining the suitability of the test method, and although limited in scope, do suggest the following trends which may be worthy of note:

   (a) At 50 percent rh, maximum odor adsorption during a 24-hr period occurs at about 75 F, and is considerably less at 60 F and 90 F.

   (b) At 75 F, odor adsorption during a 24-hr period is greater at 50 percent than at 85 percent rh.

Further work will be necessary to confirm these preliminary findings.

REFERENCES

1. Standard Method for Measurement of Odor in Atmosphere (Dilution Method) (*American Society for Testing Materials Designation* D 1391-57).
2. ASHAE Research Report No. 1587—Temperature and Humidity Effect on Odor Perception, by W. F. Kerka and C. M. Humphreys (ASHAE Transactions, Vol. 62, 1956, p. 531).

Research program will be expanded greatly

(*Continued from page* 91)

will continue its important work as an appointed Committee of ASHRAE.

The Technical Advisory Committee on Combustion is carrying forward its plans for having an exploratory conference at the Laboratory to delineate the most pressing combustion problems requiring research for our industry. It has been found necessary to advance the date of this conference from March, when it was originally scheduled, to a more suitable day during the week of May 17. Announcements will be made to the membership concerning this as soon as further plans have been crystallized.

At the meeting of the Technical Advisory Committee on Odors, a report was made on the odor programs which have just been completed in the odor test room, and an active plan of action was laid out for the immediate future. Before additional testing can be started, however, it was deemed necessary to re-survey present odor-control methods. This study will explore the methods which are currently available either by adsorption processes, by use of odor counteractants and by masking agents. In addition to this, some thought will be given to any possibility of odor control by means of combustion methods. Odor control is of extreme importance to our whole industry as increasing evidence is coming to light that the widely-used method of air dilution as the main means of controlling odors has severe limitations. As the research program gets under way, it will be necessary to develop standardized odorants which can be used for reproducing the various types of common objectionable odors met in air conditioned spaces.

PRESIDENT'S PAGE

The real meaning of ASHRAE

CECIL BOLING

Frankly, I am not greatly impressed by numbers. One hundred million people can be as wrong as they could be right or vice versa. If numbers alone counted India and China would be of greater significance in the world of today than is our own country.

ASHRAE is far more than the nearly twenty thousand people engaged in a more or less common activity using closely related engineering principles for the consummation of their individual goals.

First, ASHRAE is opportunity – to improve upon things as they are, to broaden the horizons of accomplishment and to insure more efficient and economical use of materials, energies, resources and efforts.

Second, it is figuratively a larger and more easily opened door for the man who has chosen engineering as a career and heating, refrigerating and air conditioning or ventilating as his field for activity.

Third, it is a challenge that we make the most of the inherent possibilities created by this now more broadly enveloping professional society.

Perhaps that sounds rather idealistic; you might have hoped that I would spell out in terms of specific things that would be personally rewarding to you. I do not think that to be the way that a professional society should function nor the way that an engineer should approach his membership in such an organization.

It requires no great study to realize that over the longer interval we must give in order to get and that, in the final sense, the more we give the more we get.

So let us approach the future of ASHRAE with the idea of how much we shall contribute as individuals to the possible progress of our Society, of our companies, of our associates and of ourselves.

Many of us know of the long hours put into the development of Standards, the furthering of the work of committees, the preparation of meetings papers and other lesser chores which may seem to go unrewarded and little noted. I do not think that they are ever purposeless or lacking in true effectiveness. The question is did we give enough to make the activities worthwhile?

As the first president of the American Society of Heating, Refrigerating and Air Conditioning Engineers, I feel that my part presently is to expedite organizational phases of the new Society for the sooner we establish ASHRAE as an operating entity the sooner we may proceed to those broader accomplishments which I envision for 1959 and later years. Other days will present other problems but if we have built well the future will be all that you and I have hoped in even our most optimistic moments.

What ASHRAE Chapters are doing

Acoustics, insulation and hot water heating headed the wide range of technical topics at local meetings. Some combined chapters and sections, and other groups meeting separately, covered subjects from food storage to solar energy.

DAYTON . . . Members of both societies planned to meet jointly in February to hear Thom Muir, publishing director with Industrial Publishing Corporation, on the View from the Top of a Masthead.

ASRE men were addressed by Warren Spofford, heating and cooling systems engineer, General Electric Company, in January. With his subject, Design and Application of Residential and Commercial Heat Pumps, he discussed the basic design consideration for air to air heat pumps.

He showed curves of output vs. input to the heat pump under varying outdoor temperatures and explained how the heat pump capacity could be balanced against the heat loss characteristics of the conditioned space.

He covered briefly the special controls required for successful heat pump operation, including defrost controls, two-stage heating room thermostats, and supplementary electric heat.

LOS ANGELES (R) . . . Dehumidification is coming of age, said C. J. Otterholm, engineer for Richard Dawson Company, partly due to the extensive missile and rocket propellent development program.

Speaking at the January meeting, he explained that some applications are more economical using solid or liquid desiccants, but refrigerated dehumidification is better suited for most jobs. Missile preflight dehumidification puts strict demands on the refrigerated system, he said; the refrigeration plant must operate automatically from no load to full load continuously.

Air to air heat exchangers are used in some systems, he continued, to obtain up to 40% sensible heat recovery. The heat exchangers are constructed of 0.025 in. ½ hard aluminum, operating with a velocity of 1000 fpm.

ARIZONA (H) . . . L. S. Wirt, engineering specialist, acoustics, AiResearch Manufacturing Div of Garrett Corporation, presented a talk on Sound recently. He covered problems encountered specifically in the air conditioning industry.

AUSTIN (H) . . . Electrostatic Air Filters were the subject of a talk by W. P. Dickson, senior sales engineer, (Dallas) Minneapolis-Honeywell Regulator Company, when he spoke to members of this chapter recently.

At an earlier meeting H. E. Straub, research engineer, Titus Manufacturing Corporation, gave facts on air distribution, in which items such as primary air, total air, concept of a stagnant layer, natural convection currents, return intake and room air all play an important part. One of his conclusions was that any single supply outlet will not effectively suffice for both heating and cooling.

CLEVELAND . . . The wide acceptance of high-velocity air distribution as a practical and economical system should spur the engineer to learn all he can about it, said Elliot Godes, Anemostat Corporation, at a combined meeting of the two societies in February.

A successful high-velocity duct system must be simple, complete and flexible, he stressed.

In January, at the last regular ASRE meeting, George Sinichko, Dunham-Bush Company, spoke on Air Cooled Condensers in use in Air Conditioning.

DETROIT (R) . . . Two men in the automotive industry planned to meet members of this section in February to discuss Automobile Air Conditioning.

J. D. Loveley, in charge of direction of activities involving design, development, testing and releasing all vehicle air conditioning units at Chrysler Corporation, and Russell D. Foley, Automotive Div air conditioning engineer, American Motors Corporation, were the guest speakers.

SOUTHWEST TEXAS (H) . . . J. P. Licandro, district chief engineer, Carrier Corporation (Dallas) presented a talk to members on the subject of Absorption Refrigeration. Using slides, he traced the cycle of lithium bromide involving balancing of pressure and temperature. He

CHAPTER AND SECTION DATES*

	Mar.	Apr.	May
Atlanta (H)	9	13	11
Austin (H)	19	16	21
Arizona (H)	2	6	4
Arkansas (H)	19	21	19
Baltimore (H)	18	15	20
Baltimore (R)	10	2	7
Baton Rouge (H)	18	15	20
Birmingham (R)	18	18	14
Bluegrass (H)	9	13	11
British Columbia (H)	11	8	13
Boston (R)	3	7	5
Buffalo (R)	5	—	7
Carolina (R)	13	10	8
Central Michigan (R)	19	16	21
Central New York (H)	18	15	20
Central New York (R)	18	15	13
Central Ohio (H)	16	20	18
Central Pennsylvania (R)	10	8	6
Chicago (R)	12	9	14
Cincinnati (H)	3	7	5
Cincinnati (R)	16	20	—
Cleveland (R)	2	4	2
Colorado (R)	9	13	11
Columbus (R)	23	27	25
Connecticut (H)	19	16	21
Connecticut Valley (R)	12	9	14
Dallas-Fort Worth (R)	9	13	11
Dayton (R)	10	14	23
Delta (H)	17	21	19
Detroit (R)	9	6	4
El Paso (H)	16	20	18
Empire State Capital (H)	11	8	13
Erie (R)	—	14	10
Evansville (R)	3	7	5
Florida West Coast (R)	—	—	—
Fort Worth (H)	18	15	20
Georgia (R)	9	13	11
Golden Gate (H)	5	2	7
Hudson-Mohawk (R)	16	20	18
Houston (R)	3	7	5
Illinois (H)	9	13	11
Illinois-Iowa (H)	16	20	18
Indiana (H)	24	28	26
Indianapolis (R)	18	15	20
Inland Empire (H)	2	6	4
Iowa (H)	10	14	13
Jacksonville (H)	3	7	5
Johnstown (H)	—	—	—
Kansas (H)	3	7	5
Kansas City (H)	2	6	4
Kansas City (R)	16	20	18
Long Island (H)	9	13	11
Los Angeles (R)	—	—	—
Louisville (R)	23	27	25
Manitoba (H)	26	23	25
Massachusetts (H)	17	21	19
Memphis (H)	16	20	18

* Pending establishment of unified ASHRAE chapters, meetings of former ASHAE Chapters (H) and former ASRE Sections (R) are proposed upon the basis indicated above for early months.

included the generator, condenser and absorber with the heat exchanger in the cycle.

Pointing out the advantages and disadvantages of steam vs. electric power, he also compared initial and operating costs and related factors.

INDIANAPOLIS (R) . . . Debate on hermetics and semi-hermetics vs. open belt type compressor units was the feature of the February meeting. A. R. Seebach, Jr., and John Snellenberger were on the side of hermetics, while William Friege, Jr., and James Smith spoke on the better points of open type units.

ATLANTA & GEORGIA . . . Meeting jointly in February, members planned to hear Dr. J. G. Woodroof, Chairman of the Div of Food Technology, University of Georgia, on the Refrigeration of Foods.

BALTIMORE (H) . . . R. J. Caffrey, sales engineer, Johnson Service Company, delivered a talk to this chapter recently on Pneumatic Control Centers.

BIRMINGHAM (R) . . . Fans and their Practical Application were on the agenda for the January meeting of this section. H. D. Diffenbaugh, manager, Fan Department, American Blower Div of American-Standard, was the guest lecturer.

KANSAS CITY (H) . . . Dividing his talk into two parts, M. E. Coobs, Johnson Service Company, dealt first with the control of centrifugal and reciprocating refrigeration machines and then covered the control of cooling coils. His lecture was entitled Refrigeration Controls as Applied to Air Conditioning.

CHICAGO (R) . . . Three talks in all were planned for the mid-February meeting. Before dinner members heard W. J. Monkerud, Chief of the Service Department, Air Comfort Corporation, on a Service Department's Function. He covered its organization, principles, aims and customer service.

After dinner, two programs ran simultaneously. At one, Otto J. Nussbaum, Chief Engineer, Kramer-Trenton Company, covered the Application Merits of Air Cooled Condensers, Cooling Towers and Evaporative Condensers.

At the other, Mrs. Florence B. Anderson, Norge Div, Borg-Warner Corporation, discussed Future Design Trends of Household Refrigerated Products.

NEBRASKA (H) . . . Meeting recently, this chapter heard T. W. Peterson on Temperature Controls with Hot Water. He is sales engineer, Barber-Colman Company.

LONG ISLAND (H) . . . Vice president, Sarcotherm Controls, Inc., T. R. Stevens met with members of this chapter recently to discuss hot water heating, application, design and installation.

LOUISVILLE . . . The art of pouring foamed in place insulations, as well as other applications of foam insulations were planned for discussion at the January meeting. Dr. Robert W. Finholt, Manager of Chemistry Laboratory, Appliance Park, was the guest speaker.

EVANSVILLE (R) . . . Noting that Refrigerant Blown Urethane Foams are of particular interest in refrigeration because of the low K factor which permits thinner insulation, J. J. Gorman, technical sales rep resentative for the Elastomer Chemicals Department, E. I. du Pont de Nemours & Company, chose them as his topic at the February meeting.

JACKSONVILLE (H) . . . Package Generators were the subject of a talk given by W. T. Gardner, Cyclotherm Div, National-U.S. Radiator Corporation before members of this chapter.

ILLINOIS (H) . . . Using slides to illustrate air distribution in its better forms, Peter Van Dae, manager, sales promotion, Barber-Colman Company, addressed chapter members recently. Air outlets actually are functional now and no longer are to be considered merely items to be treated artistically, he said.

Air distribution is more noticeable in winter than in summer, he went on, because of the tendency of people to complain of drafts. A classroom unit ventilator air supply may vary from 55 to 90 F; it is air delivery that is important here.

Several important factors which he stressed in grille selection are temperature differential, height and length of room and architectural obstructions. He concluded his talk with a discussion of noise level measurements.

WASHINGTON, D. C. (H) . . . Many different types of air conditioning systems being designed and installed in multi-story office buildings came under the scrutiny of M. J. Wilson, product specialist, Carrier Corporation, in addressing members recently.

Generally, he said, these systems fall into the categories of air-water systems, all air systems and

(Continued on page 135)

	Mar.	Apr.	May
Miami Valley (H)	4	1	6
Michigan (H)	16	13	11
Mid South (R)	9	13	11
Milwaukee (R)	3	7	5
Minnesota (H)	2	6	4
Mississippi (H)	23	27	25
Mobile (R)	23	27	25
Montreal (H)	16	20	18
Montreal (R)	9	13	11
National Capital (R)	10	7	8
Nebraska (H)	10	14	12
New Mexico (H)	17	21	19
New Orleans (R)	10	6	4
New York (H)	16	20	18
New York (R)	2	6	4
Northeastern Oklahoma (H)	10	14	12
Northern Alberta (H)	18	15	20
Northern New Jersey (R)	—	—	—
Northern Ohio (H)	9	13	11
Northern Piedmont (H)	—	—	—
North Jersey (H)	10	14	12
North Texas (H)	16	20	18
Oklahoma (H)	9	13	11

	Mar.	Apr.	May
Ontario (H)	2	6	4
Ontario (R)	6	3	1
Oregon (H)	5	9	7
Oregon (R)	—	—	—
Ottawa Valley (H)	17	21	19
Pacific-Northwest (R)	—	—	—
Panama Canal Zone (R)	17	15	20
Philadelphia (H)	12	9	14
Philadelphia (R)	6	3	8
Pittsburgh (H)	16	20	18
Pittsburgh (R)	9	4	—
Puget Sound (H)	10	14	12
La Ville de Quebec (H)	—	—	—
Richmond (R)	2	6	4
Rochester (R)	4	4	—
Rocky Mountain (H)	4	1	6
Sacramento Valley (H)	4	1	6
San Antonio (R)	10	14	12
San Francisco (R)	19	23	21
San Joaquin (R)	17	21	19
Savannah (H)	—	—	—
Shreveport (H)	19	16	21
South Carolina (H)	16	20	18

	Mar.	Apr.	May
Southern Alberta (H)	—	—	—
Southern California (H)	3	7	4
Southern Connecticut (R)	17	21	19
Southern Piedmont (H)	30	27	25
South Florida (R)	10	14	12
South Texas (H)	20	17	15
Southwest Texas (H)	17	21	19
St. Louis (H)	11	8	13
St. Louis (R)	16	20	18
Switzerland (H)	—	—	—
Toledo (H)	2	6	4
Twin City (R)	9	13	11
Utah (H)	19	16	21
Virginia (H)	18	15	20
Washington, D. C. (H)	11	8	13
Western Canada (R)	23	27	25
Western Massachusetts (H)	—	—	—
Western Michigan (H)	9	13	11
Western Michigan (R)	4	1	6
Western New York (H)	9	13	11
West Texas (H)	3	7	5
Wichita (R)	17	21	19
Wisconsin (H)	16	20	18

People

Meetings ahead

March 1-4—National Association of Frozen Food Packers, 18th Annual Convention and Machinery and Supplies Exposition, Conrad Hilton Hotel, Chicago, Ill.

March 9-12—Aviation Conference, sponsored by the aviation div of The American Society of Mechanical Engineers, Statler Hilton Hotel, Los Angeles, Calif.

March 16-20—Western Metal Exposition & Congress, 11th Congress, Pan-Pacific Auditorium and Ambassador Hotel, Los Angeles, Calif.

March 16-20—National Association of Corrosion Engineers, National Meeting, Sherman Hotel, Chicago, Ill.

March 31-April 2—21st American Power Conference, Hotel Sherman, Chicago, Ill.

April 1-3—Gas Appliance Manufacturers Association, Annual Meeting, American Hotel, Bal Harbour, Fla.

April 2-5—The Refrigeration Research Foundation, Statler and Mayflower Hotels, Washington, D. C.

April 5-9—National Association of Refrigerated Warehouses, Statler and Mayflower Hotels, Washington, D. C.

April 5-9—Edison Electric Institute, Annual Convention, New Orleans, La.

April 29-May 4—Oil Heat Institute of America. Inc., Annual Convention, Seattle, Wash.

May 3-6—Air-Conditioning and Refrigeration Institute, Board and Annual Meeting, The Homestead, Hot Springs, Va.

May 17-20—Industrial Heating Equipment Association, Inc., The Homestead, Hot Springs, Va.

May 25-28—Design Engineering Show, 4th Annual Exposition and Conference, sponsored by the machine design division of American Society of Mechanical Engineers, Philadelphia, Pa.

June 11-13—Heat Transfer and Fluid Mechanics Institute, University of California, Los Angeles, Calif.

June 22-24—American Society of Heating, Refrigerating and Air Conditioning Engineers, first annual meeting, Lake Placid, N. Y.

T. W. McLorg has been selected full-time executive secretary of the newly formed Canadian Refrigeration and Air Conditioning Association, a composite of manufacturers, contractors and wholesalers. ASHRAE members elected to office at the late January meeting included **I. M. Bodine,** general sales manager, Keeprite Refrigeration Ltd., as president; **John B. Graydon,** general manager, Robert Elder, Ltd., 1st vice president; and **Walter Smallwood,** owner, Smallwood Equipment Sales, 2nd vice president. Chosen for the CRACA National Board of Directors were **J. F. Gibbs,** Refrigeration Supply; **W. J. A. Marshall,** Marshall Refrigeration Company; **E. G. Pullen,** Air Coils Manufacturing Company; **L. Milligan,** district manager, Robbins & Myers Company; **J. C. E. Lowcock,** deputy div manager, Linde Hall Div of J. & E. Hall; and **H. G. S. Murray,** president, Murray Air Conditioning Ltd. Only companies or individual operating as companies are eligible for membership in the new organization.

Norman F. T. Saunders becomes managing director of Kelvinator, Ltd., a British subsidiary of the American firm, after twenty years as general manager of Coldair, Ltd., refrigeration products subsidiary of General Electric Company, Ltd. (of England). An engineering graduate of the University of Glasgow, he has written several technical papers, and a book on Factory Organization and Management. In his new capacity he will head the new manufacturing facilities of Kelvinator at the Bromborough, England, location.

Walter R. Yeary has been named manager, residential projects engineering, air conditioning department, General Electric Company, in Tyler, Texas. He is the author of numerous technical papers and magazine articles on heating and air conditioning applications.

Dennis Timmons will act as an application engineer at the Dunham-Bush, Inc., plant in Riverside, Calif., and as sales engineer in the Riverside area in his new post with that firm.

Fred G. Peck has been chosen vice president–engineering by the Ranney Refrigerator Company. Formerly employed by Victor Products Corporation, he joined his present firm in 1952 as chief engineer. He is presently an Associate Editor on the Unitary Refrigeration Equipment Section of the Refrigeration and Air Conditioning Equipment volume of the ASHRAE Data Book planned for later publication. He also served on the ASRE General Technical Committee.

J. M. Lizotte has been appointed manager of the Trane Company of Canada sales office in Quebec City and district. He has been sales engineer with the firm in all fields of air conditioning, heating and ventilating.

F. E. Dennison retires as manager of quality control and inspection, York Div, Borg-Warner Corporation.

William V. Mauer, president, Custom-Temp Corporation, Industrial Manufacturers, has been elected president of the New York State School Boards Association.

Charles J. Moll is covering the southwestern region as sales manager with Westinghouse Electric Corporation. He will headquarter in St. Louis.

Robert Allan moves to Detroit headquarters where he will be sales representative for Wolverine Tube, Div of Calumet & Hecla, Inc.

Russell H. Burgess has been elected vice president in charge of the air conditioning and cool-heat departments of Drying Systems Company, a div of Thor Power Tool Company.

Others are saying—

that . . . . . where and how to purge non-condensable gas from ammonia systems is not an agreed upon procedure. Authors D. D. Wile and T. H. Rea approach the problem from the somewhat different angles of points of non-condensables concentration and trap performance and connections. *Industrial Refrigeration, January 1959, p 12.*

that . . . . . based upon a generator with a linear speed/voltage characteristic delivering 12 volts at maximum speed of 3000 rpm a remote-reading vane anemometer for medium velocities has been developed. This device has a torque opposing rotation of less than 1 grm/cm. A rotor, 4 in. diam and fitted with eight blades, has been mounted directly on to the generator shaft, and the vanes are protected by a brass band which is supported on the casing of the generator. The upper limit of air speed which may be measured with this type of anemometer is dependent upon the upper limit of rotational speed for which the generator is rated (here, 3000 rpm). *Journal of The Institute of Heating and Ventilating Engineers, January 1959, p 278.*

that . . . . . more and more attention is being focused on control of rising noise levels in the home. While the decibel as a unit of measure has unlimited applications in acoustics, the lay tendency to mis-use it or interpret it as a valid single number measure of subjective reaction to noise creates many problems. A more suitable approach is to translate the results of a noise evaluation into terms of predicted level of consumer acceptance. *Noise Control, January 1959, p 48.*

that . . . . . six basic systems of air conditioning for ships include zone control, room volume control, terminal reheat, air-water induction, dual duct and dual conduit. It is impossible to claim that any one system is best for all ships, because of the wide variations in structure, performance requirements and economic factors. *Marine Engineering/Log February 1959, p 62.*

that . . . . . conditioning of rooms by means of air displacement is advantageous only in the case of a great air exchange rate. Results of experiments with air jets may be correlated by means of simple mathematical equations which serve as a basis for further calculations. The behavior of a single air jet in a limited space has been investigated, allowing for an estimation of the depth of penetration of a plane jet which enters the air space in the vicinity of the ceiling. *Kaltetechnik, January 1959, p 3.*

that . . . . . cold storage for cut dried fruits is recommended to eliminate the problem of deterioration and loss of color occurring in the time between field drying and marketing. Formerly, ordinary temperature storage resulted in an inferior looking product. *Canner/Freezer, February 1959, p 2.*

that . . . . . three refrigerated sea water tanks in the 45 ft Canadian salmon troller *Ruth G II* are in a set across the vessel, allowing a holding of 17,000 lb. 30,000 lb more may be carried in the conventional hold. This spacing of the three tanks permits running in three different conditions of loading without ballasting with another tank. Refrigeration is supplied by a high-speed automotive-type compressor, belt driven from the main engine and controlled as required. *Progress Reports of the Pacific Coast Stations (Canadian), No. 112, December 1958. p 3.*

Carl F. Kayan, professor of Mechanical Engineering, Columbia University, received one of the Great Teacher Awards from that university's Society of Older Graduates. Columbia College and School of Engineering graduates who received degrees at least 30 years ago, and who have continued to give service to the university comprise the members. His citation read in part " . . . honored . . . for his research and writing, particularly in the important areas of heat-transfer, air conditioning and refrigeration, he is honored most by the place he earns in the hearts of those he teaches . . . "

Merlin J. Holst will be responsible for OEM sales within the refrigeration industry in his new position as Sales Manager, Refrigeration Products, Wall Tube and Metal Products Company. A graduate of the University of Michigan Engineering College, he has been active in many phases of management with the firm for the past 10 years.

Robert Carow, owner of Bob Carow Company in Lubbock, Texas, has been named sales engineer in the Texas Panhandle territory of Acme Industries, Inc.

Hal K. Jennings, for the past 10 years vice president of Kuempel Company, is now director of Keco Industries, Inc. new commercial and industrial contracts div.

A. J. Lane, District Manager (Atlantic Central area) for Dole Refrigerating Company, was elected International Director of the Refrigeration Service Engineers Society. In his position with the society he will represent the Carolinas, Georgia and Florida.

Roy G. Trafton becomes associated with the refrigeration div of McIntire Company as district representative for its line of DFN driers, filters and strainers. Formerly with Alco Valve Company, he will cover Arkansas, Louisiana, Mississippi and western Tennessee.

Joseph S. Michener will work from Edmonton in his new appointment as western district manager for Dunham-Bush of Canada, Ltd.

Stewart E. Lauer retires as chairman of the board of York Corporation (a subsidiary of Borg-Warner Corporation), and as chairman of the supervisory board of the York Div of Borg-Warner Corporation. He has been with the firm for nearly 48 years. A mechanical engineering graduate, University of Pennsylvania, he was recently the recipient of an honorary degree from Ursinus College for his outstanding work as engineer, industrialist and citizen. Mr. Lauer will remain active in a consulting capacity.

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- ☐ **C-1** Pressure-Enthalpy Diagrams for All "Freon" Refrigerants
- ☐ **C-4** Pressure-Temperature Relationship Charts of Refrigerants
- ☐ **C-14** Viscosity Charts of "Freon-12" — Oil Solutions
- ☐ **C-15** Solubility Chart of "Freon-11" in 300 Viscosity Oil
  Chart of Percent by Weight "Freon-12" Absorbed in Mineral Oil (325 Viscosity)
  Chart of Percent by Weight "Freon-12" Absorbed in Mineral Oil (150 Viscosity)
- ☐ **C-17** Entropy/Enthalpy Chart for "Freon-12"
- ☐ **C-18** Heat Capacity Chart of "Freon-12"
- ☐ **M-5** Solubility of Water in "Freon" Liquid
- ☐ **M-14** Calculations of the Velocity of Sound in "Freon" Vapors
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- ☐ **RT-9** The "Freon" Family
- ☐ **RT-11** Design Factors When Using "Freon-12" Refrigerant
- ☐ **S-1** MH-2256 — Report of Underwriters' Laboratories on "Freon-12"
- ☐ **S-3** MH-2630 — Report of Underwriters' Laboratories on "Freon-21"
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- ☐ **S-13** Decomposition of "Freon-12" by Flames
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- ☐ **S-18** Toxicity Information on "Freon-13"
- ☐ **T-11A** Thermodynamic Properties of "Freon-11" at Low Pressures
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- ☐ **T-114B** Thermodynamic Properties of "Freon-114"
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Emphasis for Standards

A. T. BOGGS III
ASHRAE Technical Secretary

Development and promulgation of industry standards will be one of the most important activities of ASHRAE. Sec. 8.8.24 of the By-Laws charges your Standards Committee with the selection, development, and preparation of all codes and standards in the fields of heating, refrigeration, and air conditioning and ventilating engineering. The activities of the Standards Committee shall be solely for the development of engineering science and it is further charged to cooperate with other industry organizations in the development, preparation, and adoption of codes and standards.

The membership of the first Standards Committee of ASHRAE will be made up of the members of the previous Standards Committees of ASRE and ASHAE. Cooperation with similar standards groups in allied industry and government agencies will be continued on an expanded basis. In this manner it will be possible for ASHRAE standards to be of maximum benefit to those using them.

It is planned that close liaison will be maintained between the subcommittees developing ASHRAE standards and all groups who may have an interest in these standards. Many Society members are interested in the development of industry standards and in order that this group may have a source of information on existing industry standards concerning the areas of interest in ASHRAE, there will be a monthly column in the ASHRAE **Journal** reviewing new or revised standards.

Specific questions with respect to any of the standards reviewed may be submitted to the technical secretary. A similar review space covering standards in the refrigeration and air conditioning fields was included as a part of **Refrigerating Engineering.** During 1958 the previous column reported on 66 industry standards or reports concerned with standards. Eleven national bodies developing standards for industry were included and it is planned that the established contacts with these groups will be maintained. With the wider scope of responsibility for ASHRAE standards activities, it will be necessary to expand this liaison activity so that standards covering the entire temperature scale may be brought to the attention of society members. For those members who have interests in national and international aspects of industry standards, the development and availability of standards in these areas will be included.

A technical society will progress and serve the industry in direct proportion to the willingness of the members to serve on national and local committees. ASHRAE has a tremendous wealth of engineering talent covering all aspects of heating, refrigeration, and air conditioning. If any member is particularly interested in standards and would be willing to serve on subcommittees developing Society standards, it is requested that such interest be brought to the attention of the Standards Committee.

ARI Certification Program: The first edition of the Directory of Certified Unitary Air Conditioners was made available during the first week of January. This report is a result of the program initiated several months ago by ARI in cooperation with National Warm Air Heating and Air Conditioning Assn. The directory is being made available to the public through dealer and contractor outlets of the 31 participating companies, which produce more than 80% of all the unitary equipment manufactured in the U. S. Copies will also be available from Better Business Bureaus and direct from ARI. The directory lists the Btuh ratings of the unitary equipment produced by 20 of the 31 cooperating companies. Data on the others will be included in subsequent supplements to be issued on a monthly basis, as required. The certification program is based on firm, enforceable contracts, under which the seal of certification may be withdrawn if equipment does not meet the claimed ratings. It is not a "voluntary" program, once a contract to participate has been signed.

Foreign standards: The American Standards Association (ASA) is the national clearinghouse and coordinating body for standardization in the U. S. and represents American interests in international standards work. ASA has available English translations of several German and

(*Continued on page* 108)

The unfolding atom and space age with its unprecedented scientific discovery and technological development calls for corresponding advances in standards and their application to science, industry, commerce and trade. The responsibility for progress does not rest alone with the National Bureau of Standards – it must be shared by everyone concerned with standards.—A. T. McPherson, Associate Director, National Bureau of Standards, so addressed the Southern Weights and Measures Association on October 28, 1958.

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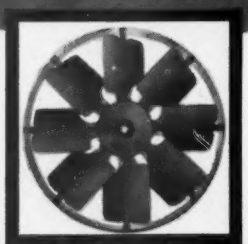

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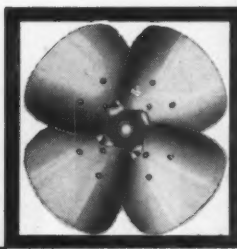

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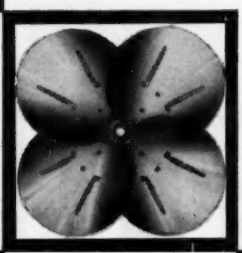

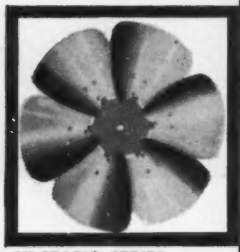

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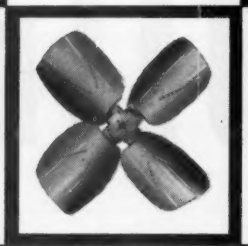

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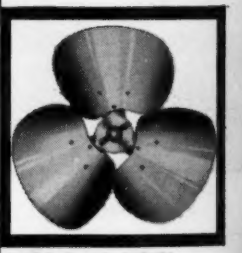

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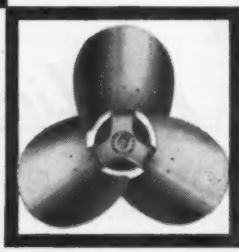

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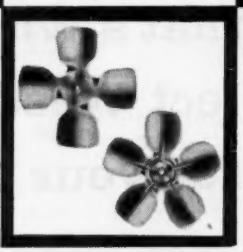

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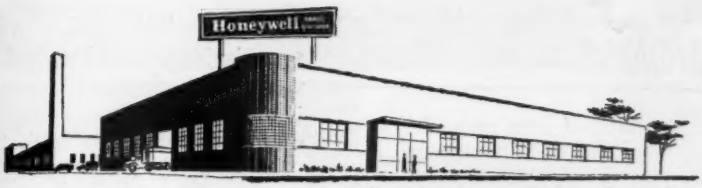

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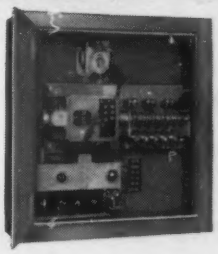

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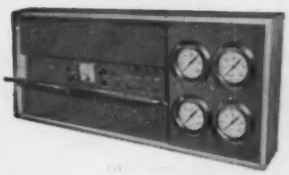

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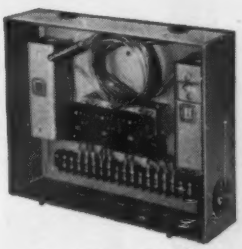

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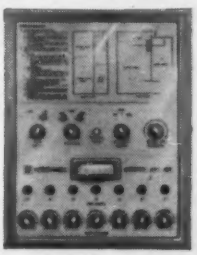

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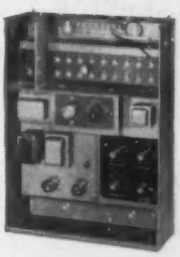

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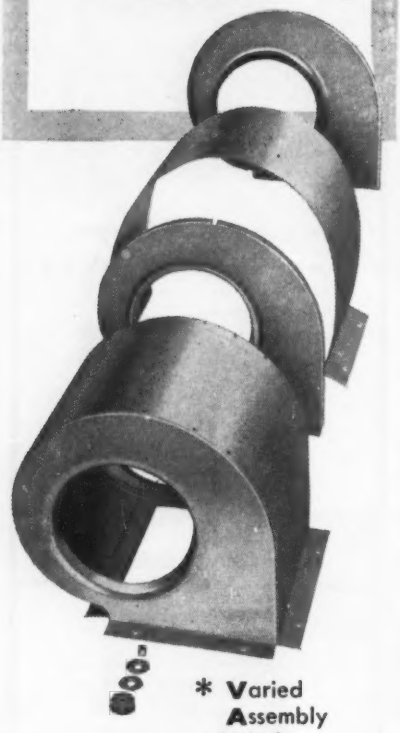

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STANDARDS

(*Continued from page* 104)

French industrial standards. The German standards cover specifications including screws, washers, nuts, bolts, test methods for steel and various nonferrous metals, welding, plumbing fixtures, and inch-millimeter conversion. Among the subjects covered in French standards are zinc alloy ingots, blast furnace cements, and conveyor belting. The complete list of the available standards as well as the individual standards can be ordered from ASA, 70 E 45th Street.

Lock Washers: The revision of American Standard B27.1, Lock Washers, has been approved by ASA and published by ASME. This standard is sponsored by SAE and ASME. It covers the dimensions, physical properties, and methods of testing of spring lock washers in various materials; carbon steel tooth lock washers; spring lock washers and machine screw assemblies; and carbon steel tooth lock-washer-and-machine-screw assemblies. Copies are available at $2 each from either ASME or ASA.

Sound Insulation: The National Bureau of Standards has announced the availability of a second supplement to the NBS Building Materials and Structures Report 144. This supplement covers sound insulation of wall, floor, and door construction and may be obtained from the Superintendent of Documents, Washington 25, at 10c per copy. This supplement contains sound insulation data for building structures measured at NBS during the period July 1955 to December 1956. The accuracy of the figures is discussed, and details are given of a new average figure, called the Energy Average, for the overall sound insulation of a panel. Its superiority to the Decibel Average which it is designed to supersede, is discussed. Measurements were made in accordance with American Standard Z24.10-1957 with the exception that the average sound transmission loss values are based on 11 frequencies rather than 9. The original BMSR 144 is available from the Superintendent of Documents for 40c and the first supplement is available at 5c.

Thermometer Calibration: The National Bureau of Standards recently announced the availability of NBS Circular 600 covering calibration of liquid-in-glass thermometers. Copies are available from the Superintendent of Documents, Washington 25, at 20c each. This circular gives information needed for the efficient use of liquid-in-glass thermometers. The methods of calibrating thermometers will be of interest to those who calibrate their own instruments as well as those who submit them to the National Bureau of Standards for calibration. The liquid-in-glass thermometer is probably the most widely used device for measuring temperature, but it is not entirely foolproof. If used with awareness of its inherent limitations, it can give excellent service where reliable measurements are desired. In the circular consideration is given to various factors that affect the use of the more common types of thermometers: total-immersion, partial-immersion, low-temperature, Beckmann, calorimetric, and clinical. Details are presented on the calculation of emergent stem corrections; and there are discussions of sources of error such as thermometer lag, gradual changes in the glass, and changes in bulb volume due to annealing. The techniques and equipment are described and information provided on the eligibility requirements and application for the Bureau's calibration services.

Plans ahead for more extended programs in various activities are discussed elsewhere in this issue of the ASHRAE JOURNAL.

With President Boling—page 97. Research Planning—page 91. Standards Program—page 104. Officers, Committees—page 102. ASHRAE Facts and Data—page 76.

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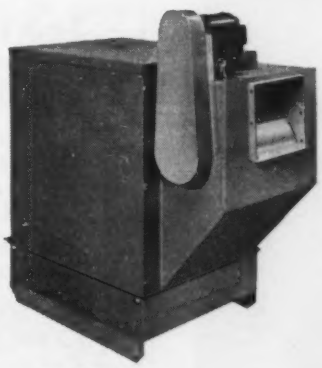
- Gravity-type distribution basin—low pumping head over tower.
- Water outlet in sump has large strainer and anti-cavitation plate, easily removed for cleaning.



CAPACITY DATA* 3 GPM/TON

Model	*Wet Bulb °F / Inlet Water °F / Outlet Water °F*	78° / 93 / 85
2 WS	TONS	2.0
3 WS	TONS	3.1
5 WS	TONS	5.1
8 WS	TONS	8.1
10 WS	TONS	10.0
16 WS	TONS	16.1
20 WS	TONS	20.0
25 WS	TONS	25.1
30 WS	TONS	32.2
40 WS	TONS	40.0
50 WS	TONS	50.2

*Based on Tower Rejection of 250 BTU/MIN/TON

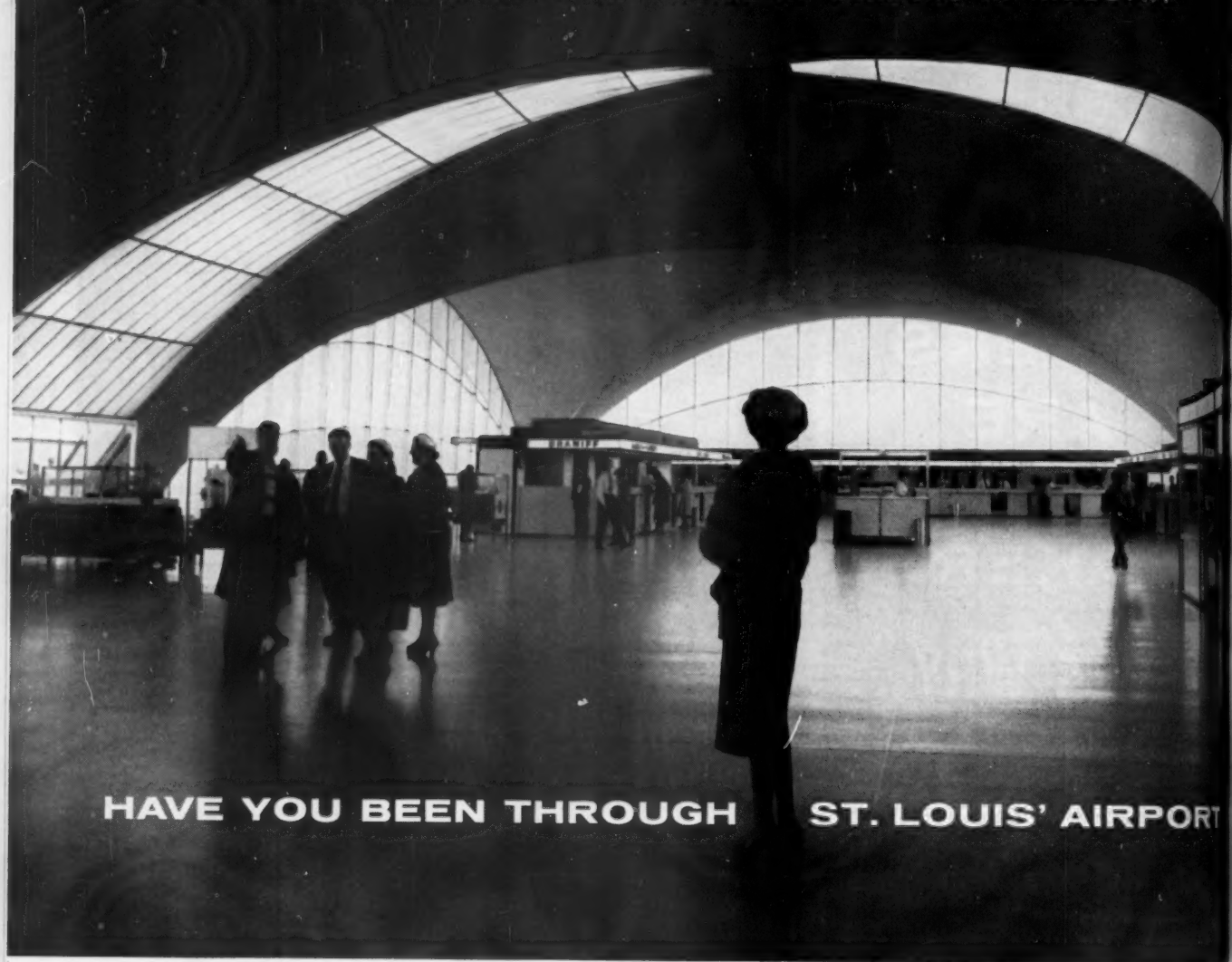
See Your Wholesaler
Or write for Bulletin 1030-D

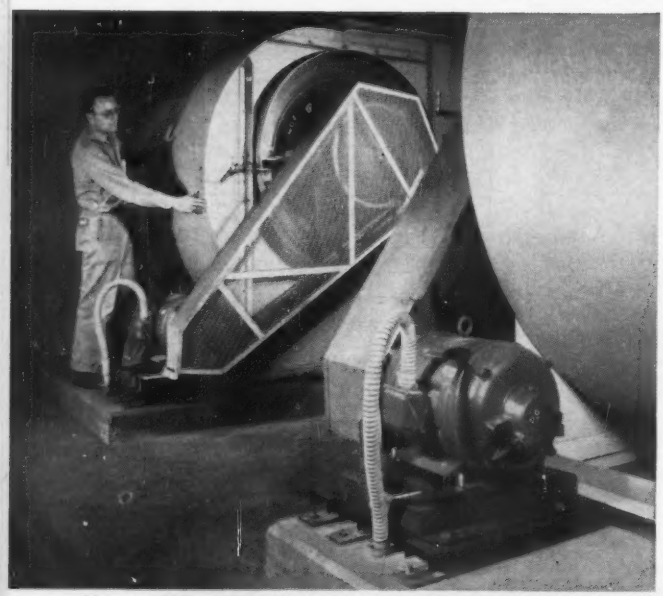
WATCHDOG OF THE NATION'S FOOD AND COMFORT ☆ LARKIN ☆

LARKIN COILS INC.

519 Memorial Drive, S.E. • P. O. Box 1699 • MUrray 8-3171
ATLANTA 1, GEORGIA

WAGNER ELECTRIC MOTORS . . . THE CHOICE OF LEADERS IN INDUSTRY





Here are two of the five air return fans, each powered with 15 hp Wagner Motors.

Chances are, you have . . . some 133,000 air travelers pass through this handsome Municipal Airport each month. And, chances are, you've been impressed with the clean, soaring architectural lines of the building and with the clean, fresh, always comfortable atmosphere.

The job of efficient air movement for this 50,000 square foot building is handled by five air supply fans and five air return fans—all driven by Wagner electric motors . . . five 40 hp motors for air supply and five 15 hp motors for the air return fans. Their quiet, efficient operation in this application is typical of the service record of Wagner motors in millions of applications all over the nation.

Whatever your motor requirements may be, Wagner can supply a standard motor or build a special motor to fit your need. Constant research and development have kept Wagner up front in electric motor design for more than 65 years—made the name Wagner one you can trust in choosing motors for your plant or for your product. Call your nearby Wagner field engineer for an engineering analysis of your next motor application. There are Wagner branches in 32 principal cities.

BRANCHES AND DISTRIBUTORS IN ALL PRINCIPAL CITIES WH59-2

Wagner Electric Corporation

6379 Plymouth Ave., St. Louis 14, Missouri.

SERVING 2 GREAT GROWTH INDUSTRIES . . . *ELECTRICAL . . . AUTOMOTIVE*

PARTS AND PRODUCTS

CEILING, WALL DIFFUSERS

Square and rectangular types of diffusers are in one, two, three and four way blows with six frame designs to fit all types of ceiling applications. The square sizes in any blow are available with round necks.

Unipattern vanes insure uniform air pattern for all designs. All inner vane sections may be changed from one blow to any other. An in-line diffuser is also available in one or two way blow.

Carnes Corporation, Verona, Wisc.

COMPACT CHILLERS

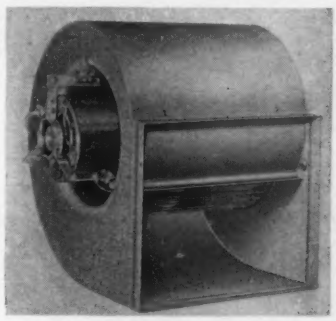
Packaged air and water cooled chillers in five sizes up to 10 ton meet a wide range of industrial requirements, including the cooling of manufacturing and processing equipment, chilling and controlling the temperature of chemicals and other liquids.

The average self-contained chiller requires 15 sq ft of floor space and may be installed either indoors or outdoors. It will maintain positive temperature control of any liquid to within 3 deg. Chillers are equipped with an oversize condenser and an easily accessible compressor with a magnetic starter. Both air and water cooled units have a self-contained recirculating water pump assembly complete with motor.

Vic Manufacturing Company, Industrial Products Div, 1313 Hawthorne Ave., Minneapolis 3, Minn.

CENTRAL SYSTEM BLOWER

Direct drive blowers which will move air at the maximum rate of 2550 cfm



at 55 in. static pressure are of 10-in. size and driven by a ½-hp motor. Minimum rate of air movement is 1000 cfm at 1.2 in. static pressure.

The blower housing measures 13⅛ x 17⅜ in. The wheel is of aluminum construction to reduce weight. It is designed primarily for use in residential central air conditioning systems.

Lau Blower Company, 2027 Home Ave., Dayton 7, Ohio.

SAFETY HEAD

Permitting the normal working pressure of a refrigerant system to be as high as 75% of the pre-determined set point of the rupture disc, this sealed type safety head eliminates the fatigue factor, giving uninterrupted service during peak cooling periods.

Black, Sivalls & Bryson, Inc., 7500 E. 12th St., Kansas City 26, Miss.

ONE PIECE HEAT PUMP

Designed for small homes and multi-unit structures, this 2 hp heat pump has a cooling capacity of 22,000 Btu and a heating output of 23,000 Btu. Additional strip heaters are 10,000 each. Automatic defrosting may be actuated every 90 min if necessary, starting only if the outdoor temperature is in the range where frost will form.

Besides the room thermostat which automatically switches the unit from heating to cooling as required,

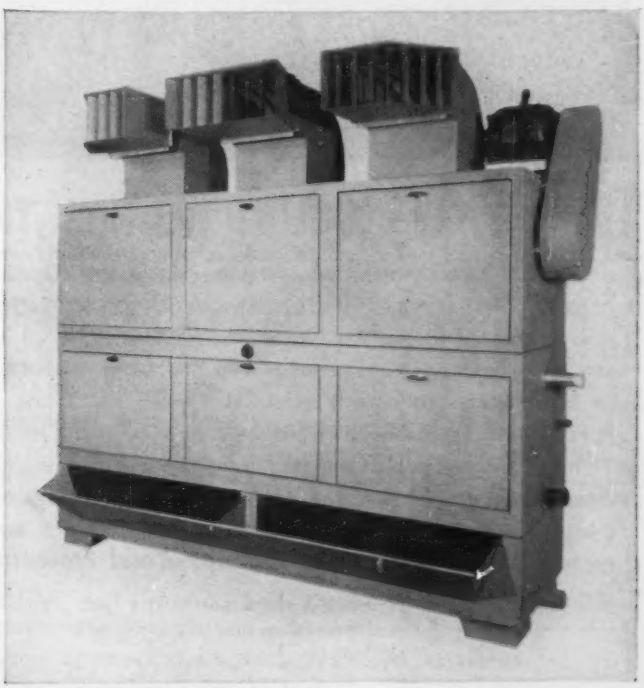
4,000—30,000 CFM COLD DIFFUSERS

for high or low temperature —hot gas or water defrost

All refrigerants can be utilized—including ammonia and brines. Coil and drain connections can be made at the right or left independent of the blower section. Blowers can be adjusted to front, rear or top discharge on the site.

Capacities: 1240 to 26,700 BTU/hr. at 1° T.D.
Total surface area: 321 to 8360 square feet.
Fin spacing: 3 or 4 per inch.
Construction: heavy 12-gauge welded steel; hot-dipped galvanized blower wheels and scrolls; hot-dipped galvanized casings available.

Send today—for your copy of free bulletin and help on installation and refrigeration problems.



KRACK ENGINEERED REFRIGERATION APPLIANCES, INC.

Manufacturers of freon, ammonia, flooded ammonia heat transfer equipment

REFRIGERATION APPLIANCES, Inc., 917 Lake St., Chicago 7, Illinois
Send free bulletin giving all technical details.
Name_____
Firm_____
Address_____
City_____ Zone____ State_____

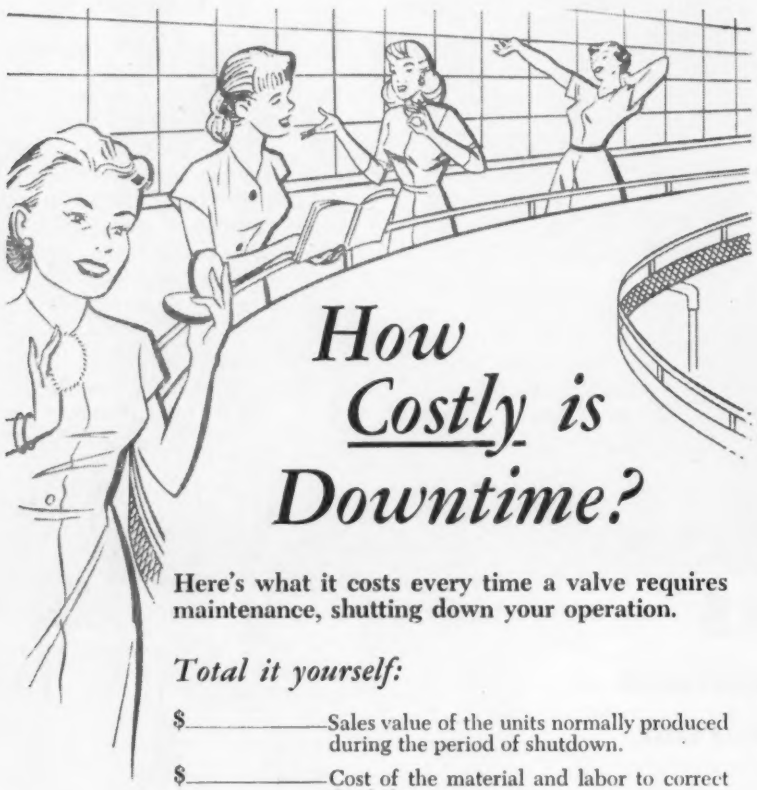
the heat pump has an outside control to actuate strip heaters when temperatures fall below certain levels. This prevents the main system from operating each time the unit demands heat.
Carrier Corporation, Syracuse 1, N.Y.

DRAFT FAN

Incorporating an advanced airfoil, backward curved blade design, this type AF Dynafoil Fan is in 13 sizes for mechanical draft and in three classes for heavy-duty applications.

The equipment has been designed to secure high efficiency and stable performance characteristics. The wheel is built with 12 aerodynamically shaped blades which in combination with the streamlined contour of the inlet permits continuously smooth flow of air through the fan. Turbulence is reduced to a minimum.

The airfoil shaped blades are of one-piece die formed construction and are welded to the rim and backplate. Internal struts inside the hollow section are provided in blades of fans for higher tip speed classes.
Clarage Fan Company, Kalamazoo, Mich.



How Costly is Downtime?

Here's what it costs every time a valve requires maintenance, shutting down your operation.

Total it yourself:

$_____ Sales value of the units normally produced during the period of shutdown.

$_____ Cost of the material and labor to correct the failure.

$_____ Idle time while operators wait for repair or work reassignment.

$_____ Added cost of supervision to "get back into production".

$_____ Cost of overhead and utilities which continue although production has stopped.

$ *Your Total Downtime Cost.*

Hubbell Valves may cost a little more, but their extra craftsmanship and dependability can save you *costly* downtime.

E. B. LaPlante
GENERAL MANAGER

HUBBELL CORPORATION
MUNDELEIN, ILLINOIS
BACK PRESSURE REGULATOR VALVES • DUAL PRESSURE REGULATOR VALVES • AUTOMATIC SUCTION STOP VALVES • SOLENOID VALVES • GAUGES • SAFETY RELIEF VALVES

"Castings to finished controls . . . every inch HUBBELL!"

ELECTRIC FURNACE

Central residential heating and cooling equipment of the Landmark series now includes an electric furnace of the convection type. Btu/hr outputs range from 68,300 to 123,000. Thermostats are two-stage, pulling in only part of the heating capacity on the first stage and all of it on the second, thus providing more constant comfort at lower cost.

A cooling coil may be placed in the cabinet at any time. These sections use single evaporators to simplify running refrigerant lines. Evaporators are positioned for low air resistance; coil fins are rippled at the outer edge.
Lennox Industries Inc., Marshalltown, Iowa.

SURFACE TEMPERATURES

For making plus or minus 2% full scale accuracy temperature measurements, this instrument is in both compensated and non-compensated types. Model DFT, as illustrated, is a compensated type for ambient temperatures in ranges 0-300, 0-450 and 0-650. The non-compensated type model ETF for relative temperature readings, is in the 0-200 range only.

There is a special 15-deg side reading probe for hard to reach surfaces. All types are self-contained and ultimate temperature response is attained in 3 to 10 sec, depending on the surface character.
Royco Instruments, Inc., 874 Fabian Way, Palo Alto, Calif.

15-80-TON COMPRESSORS

Hermetic and direct coupled drive units are included in this line of refrigeration compressors. It is in six sizes ranging from 15 to 80 ton capacities.
Worthington Corporation, Harrison, N. J.

AIR COOLED CONDENSER

Utilization of condenser heat is one of the features of this AL Aircon blower type air cooled condenser; when equipped with optional accessories, it serves the dual purpose of refrigerant condensing, as well as heating and ventilating. In seven models, with nominal capacities ranging from 10 to 50 ton, for either indoor or outdoor applications, the design incorporates forwardly curved, centrifugal fans with large outlet

TURN

TYPHOON

Adaptability of Typhoon air conditioning units is shown in this ingenious flush mounting of units at the existing windows, thus eliminating expensive duct work in the main plant of the Indiana Gear Co. in Indianapolis. 176 tons of Typhoon waterless air-cooled equipment were installed.

AND SPECIFY THE RIGHT EQUIPMENT, FOR EVERY JOB, EVERY TIME!

Now you can fill every need, solve any problem with Typhoon's exclusive commercial, industrial and residential equipment! Save time for yourself and money for your clients with Typhoon custom-built air conditioning units. You can specify the air discharge and return arrangements you need, and the most practical location for utility service and maintenance access openings. Various combinations of compressors, cooling coils and blowers in the same cabinet provide infinite variations in performance characteristics. This extreme flexibility gives you custom made units at mass production prices. A full range of air cooled units from 2 tons to 20 tons, both packaged and split systems. Water cooled units from 3 tons to 75 tons, air cooled condensing units to 20 tons, water cooled to 75 tons! And the new Typhoon Heat Pumps that open a new era in air conditioning. Each and every unit with built-in Power Reserve—a combination of superior components and oversize parts that insures dependable performance above rated capacity through all weather extremes. All copper condensers in water cooled units, largest coil surfaces and rugged 11 and 14 gauge metal cabinets. Before you plan your next system, find out how Typhoon quality and flexibility can add to your reputation for getting the job done better, faster, more economically!

TYPHOON
DIVISION OF HUPP CORPORATION

Typhoon Air Conditioning Company, Division of Hupp Corporation
505 Carroll Street, Brooklyn 15, New York

☐ Have a representative call me for an appointment.
☐ Please send full product literature.

Name_____

Firm_____

Address_____

9C-12

housings to insure quiet operation. Sixteen different unit arrangements and six different fan discharge arrangements are offered.
McQuay Inc., 1600 Broadway St., N.E., Minneapolis 13, Minn.

ENVIRONMENTAL TESTING

Refrigeration cycles designed for radiation environmental testing are incorporated in cabinets with simple plug-in application. Facilities are for tensile testing with a temperature range from −65 to 350 F. The cabinets operate with a compound type of condensing system, and combination water/air-cooled condensers. The generating equipment is surrounded by a concrete wall for safety and is operated remotely from the test chamber.
Hudson Bay Company, Div of Labline, Inc., 3070-82 W. Grand Ave., Chicago 22, Ill.

GAS HEATERS

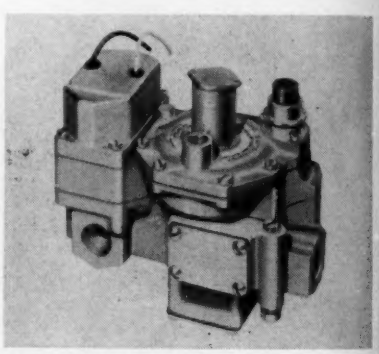
Gas-fired unit heaters in ten sizes from 25,000 to 250,000 Btu/hr incorporate a contoured aluminized-steel heat exchanger design with combustion occurring within individual tubes for maximum heat transfer.

Units are equipped with fully automatic, positive safety controls and are for any type of gas specified. Close-grained, cast-iron air adjustment shutters control the primary air to the gas flame. Adjustable louvers permit variation of discharge air to suit installation requirements.

Fan motors are standard rated at 115 volt, 60 cycle, single-phase current and are equipped with automatic reset-type thermal overload protection. Seven smaller units have permanent capacitor motors requiring periodic lubrication.
Westinghouse Electric Corporation, Sturtevant Div, Dept T-402, 200 Readville St., Hyde Park, Boston 36, Mass.

COMPACT GAS CONTROL

Covering the work of four individual controls, this single unit gas control for furnaces and unit heaters features



a pressure regulator, 100% safe shut off, safety-assured lighting and a thermomagnetic safety pilot.

The series 5250 control is an all-in-one package designed specifically to fit into gas-fired equipment without design changes. Right-hand, left-hand and straight-through outlets, with control and operator mounting positions up to 90 deg from normal make this possible.

Equipped with an electro-magnetic thermostatic operator, the unit may be used with all gases or modified for LP. A large built-in pilot filter is an accessory.
Controls Company of America, Heating and Air Conditioning Controls Div, Milwaukee, Wisc.

COMPACT CONDENSERS

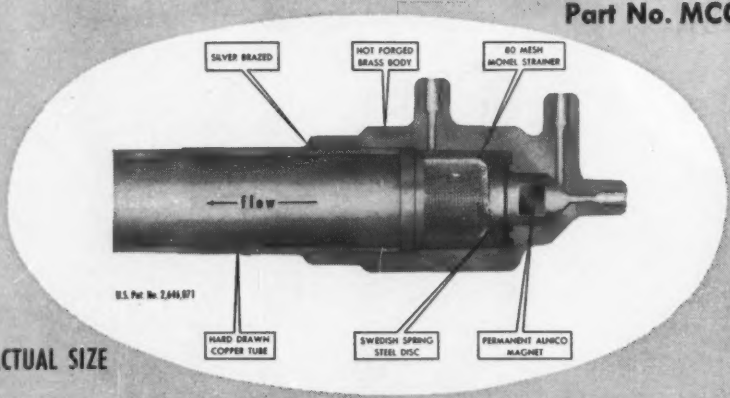
Light-weight, compact mechanical refrigeration-condensing units, incorporating greatly increased applicational versatility, are offered for cooling in aircraft galleys and related equipment. They may be mounted in confined spaces in sidewalls or under floor areas. In addition to conven-

WATSCO INTRODUCES . . .

THE NEW C.C. VALVE

(CAPILLARY CONTROL VALVE)

Part No. MCC-1



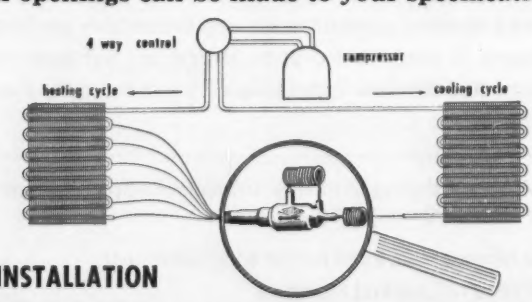
¾ ACTUAL SIZE

A valve specifically designed for use in reverse cycle systems. The *C. C. valve* is installed in conjunction with the capillaries between the high and low sides of the system, to provide single capillary restriction for the cooling cycle and double capillary restriction for the heating cycle.

The *C. C. valve* is applicable to all units up to and including five tons and may be used for either single or multiple capillary installations. Flow can be regulated by the size of the capillary used.

The *C. C. valve* replaces two ordinary check valves, thereby cutting installation time and costs in half.

Connection openings can be made to your specifications.



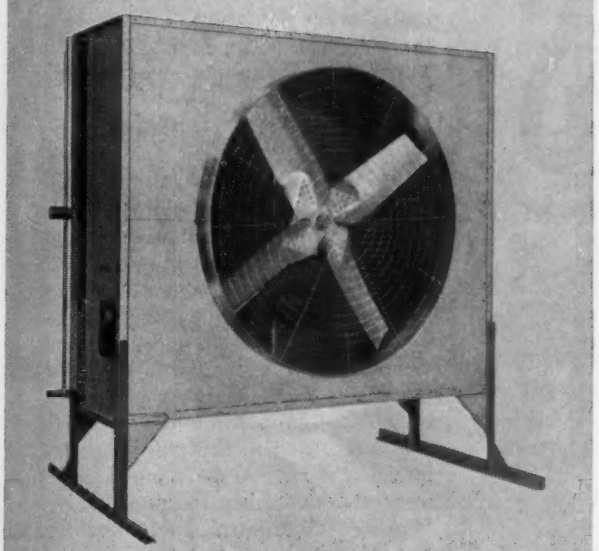
A TYPICAL INSTALLATION

Send for 1959 catalog — Dept. E-3

WATSCO INC.

1020 E. 15th STREET • HIALEAH, FLORIDA

Vertical Air-Flow Aircon



Horizontal Air-Flow Aircon

8 SIZES 7½ TO **50 TONS** IN A SINGLE UNIT

now

your choice of

VERTICAL or HORIZONTAL

AIR FLOW AIRCON CONDENSERS

Now McQuay offers you your choice of either vertical or horizontal air-flow Aircons in eight models with capacities from 7½ to 50 tons in a single unit. Unlimited tonnage is available with multiple installations. The vertical air-flow Aircon is designed to meet the needs of the architect as well as the consulting engineer. With an extremely low silhouette, the vertical Aircon is rarely visible when installed on a roof. In addition to this, the vertical air-flow Aircon is not affected by prevailing winds and therefore does not require a wind deflector.

You not only get more models and more capacity, but you get more performance, because McQuay Aircons have the exclusive McQuay Ripple-Fin coils—the finest and the standard of the industry.

The McQuay "Seasontrols," automatic head pressure controls, are available as accessories, and modulate the condenser capacity in accordance with the weather for proper operation at all times. For complete information, call the McQuay representative in or near your city, or write McQuay, Inc., 1606 Broadway St. N. E., Minneapolis 13, Minn.

McQuay Means Quality

McQuay INC.

AIR CONDITIONING • HEATING • REFRIGERATION

tional coil designs for cooler, refrigerator and freezer units, other designs feature "cold wall" construction, utilizing expansion-channel type wall-cooling plates for refrigerant circulation. Both vertical and horizontal units are in ½ and ¼ hp.
Task Corporation, 1009 E. Vermont Ave., Anaheim, Calif.

POSITIONING SWITCH

Remote control of air conditioning, heating and ventilating motors and valves is possible with a positioning switch which eliminates lag between the switch and the actuator mechanism. Twenty per cent smaller than earlier models, this switch is used primarily for remote control of pneumatic valves or motors. Turning the switch knob gradually increases or decreases the flow of compressed air to these devices, opening or closing them to the desired position. Including face plate, the switch is 2⅜ x 3⅜ x 2¼ in , in either flush or surface mounting.
Powers Regulator Company, 3434 Oakton St., Skokie, Ill.

QUENCH TANKS

Sufficient mechanical refrigeration capacity to lower the temperature of 50 lb of aluminum components from 1000 F to 34 F in 90 sec has been developed in this refrigerated water quench tank.

One self-contained unit has an air lift, top opening lid. Inside dimensions are 37 in. diam, 57 in. high. Outside cabinet dimensions are 48 x 48 in. Sufficient agitation is maintained to achieve zero stratification throughout the sweet water bath. A safety device prevents freezing.

A similar unit, designed for use as an alcohol quench tank, has sufficient mechanical refrigeration to lower the temperature of 50 lb of aluminum components from 34 to 28 F in 90 sec.

Both units maintain a temperature accuracy of plus or minus 1 F.
Webber Corporation, P. O. Box 217, Indianapolis, Ind.

NON-METAL BLOWERS

Four nylon blower wheels, for use in refrigerators and other small air-moving appliances, are light and non-corrosive. Designed for installation in standard proportioned housings with unit-bearing mounting. Diameters, in the GA series, from 2 to 3 in.
Torrington Manufacturing Company, Torrington, Conn.

CLOSET UNIT

Individual apartment control, in the heating, ventilating, filtering and humidification line, are offered in this series of compact air-handling units. In sizes from 500 to 1100 cfm, the unit is installed in a four sq ft central closet of each apartment in the building. It delivers warm or cool air to individual rooms through short ducts. Air is returned to the unit through a louvered door. A mechanical chiller may be added when required.
Hastings Air Control, Inc., Omaha 5, Neb.

VIBRATION MEASURE

Vibration frequency and amplitude measurement is one of the uses of this hand-held Vibrometer. The instrument has a reed in the form of a steel strip whose effective length may be varied by means of a knob on the side.

Also used to determine the direction of maximum transmitted disturbances, the device will locate the cause of unwanted noise due to resonant vibration and measure the speeds of rotating equipment where shafts or rotors are not readily accessible.

If the vibratory phenomenon being observed is to be displayed or permanently recorded, the output of

(Continued on page 128)

H-O-H*

GLUCOSATES*

Improved Through Research
Proven By Performance
The Best For Your
Water Treating Problems

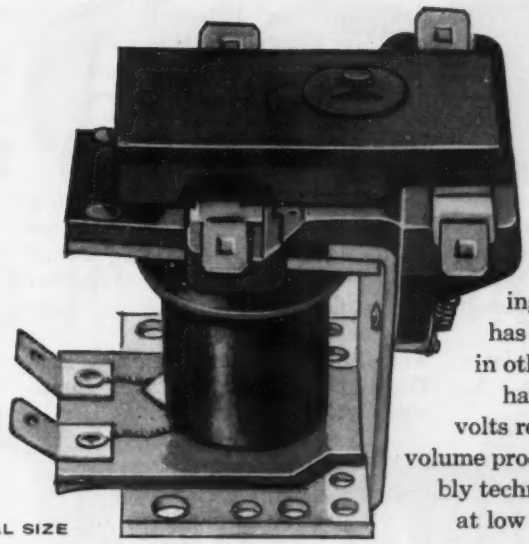
WE READ WATER* H-O-H ESTABLISHED 1931

Write for literature on your letterhead.

*REG. U.S. PAT. OFF.

D. W. HAERING & CO., INC.
ANALYSTS · CONSULTANTS · MANUFACTURERS
P. O. Box 10337
San Antonio 21, Texas

Versatile RBM Low Cost Relay

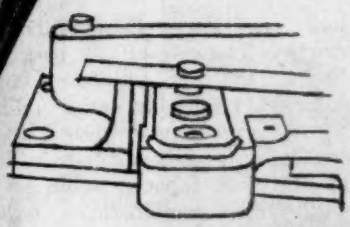


ACTUAL SIZE

ESSEX ENGINEERED

An original RBM engineering design for exacting appliance requirements that has also proven highly successful in other applications. Dependably handles up to 6000 watts at 240 volts resistive load per pole. RBM's volume production and advanced assembly techniques produce it in quantity at low cost—it's "CUSTOMERIZED."

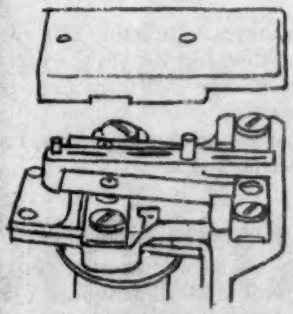
TYPE 75 POWER APPLIANCE RELAY DATA



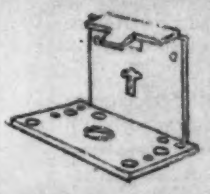
UNIQUE "HAMMER-ACTION" OPENS CONTACTS WHEN COIL IS DE-ENERGIZED

LISTED BY U/L FILE NO. 22381

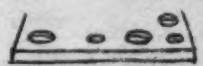
SCREW OR QUICK CONNECT CONTACT TERMINALS



MOLDED TERMINAL BLOCK, ACTUATOR AND COVER



RUGGED MAGNET FRAME



VARIETY OF MOUNTINGS

CHARACTERISTICS

- CONTACT FORM: S.P.N.O. OR 2 P.N.O.
- CONTACT RATINGS: 25 AMPS PER POLE RESISTIVE AT 240 VOLTS 60 CYCLE INDUCTIVE RATINGS – CONSULT FACTORY FOR SPECIAL INDUCTIVE RATINGS GIVING DETAILS OF APPLICATION
- COIL TERMINALS: QUICK CONNECT TYPE
- COIL RATINGS: UP TO 240 VOLTS, 50 OR 60 CYCLE (STANDARD PICK-UP VOLTS – 85% OF RATED)

Consult Your Local RBM Product Application Engineer or Write For Bulletin 1030A

RBM Controls Division

ESSEX WIRE CORPORATION, LOGANSPORT, INDIANA

Factories Located at Peru, North Manchester and Logansport, Indiana

BULLETINS

Controls. Two folders, one covering a control center designed for single-zone air conditioning systems, and the other describing a surveillance alarm system for buildings where there is critical equipment are offered with specifications and drawings.
Minneapolis-Honeywell Regulator Company, 2757 Fourth Ave., S., Minneapolis 8, Minn.

Automatic Air Filter. Construction and operating characteristics of the Model B Roll-O-Matic automatic renewable-media air filter are described in Bulletin 248-C. Included is an explanation of the 65 lineal ft roll of impregnated, reinforced media composed of continuous, slightly curled, interlaced glass filaments that are held in place by a thermo-setting plastics bond. This forms a resilient pad or fluffy blanket having a nominal thickness of two in.
American Air Filter Company, Inc., Dept PD, 215 Central Ave., Louisville 8, Ky.

Utilizing advanced design Molecular Sieve cartridges, these new Remco Filter-Driers combine unequalled drying efficiency, effective acid removal, generous flow capacity and *depth* filtration.

The massive depth filter completely removes all scale, sludge, carbon and other particles as small as 100 microns. Molecular Sieves adsorb and retain large quantities of moisture even at refrigerant temperatures of 140F, and keep moisture concentrations below 10 ppm. Acids are reduced far below dangerous corrosion limits.

Compact in size, the filter-driers are U/L Approved and may be used for Refrigerants 12 or 22, Carrene or methyl chloride. Working pressure is 500 psi; minimum bursting pressure, 2500 psi.

REPLACEABLE CARTRIDGE TYPE units use an "O" ring for a positive, leakproof flange seal. From 3 to 40 tons, with ⅜" thru 1⅝" sweat connections.

SEALED TYPE filter-driers are available in 1 to 12 tons, with ¼" thru ⅝" flare and ⅜" thru ⅞" sweat connections.

"T" FITTING TYPE in 2 to 6 tons, are readily adaptable to systems using conventional "T" driers.

Remco Molecular Sieve Filter-Driers are available at leading wholesalers. Ask your wholesaler for more information, or write for Bulletin MS-1. Remco, Inc., Zelienople, Pa.

REMCO

MANUFACTURERS OF ADVANCED REFRIGERATION PRODUCTS
Filter-Driers · Liquid Indicators · Receiver-Driers · Check Valves · Safety Devices · Frost-Tite Flare Nuts

Single Phase Motors. ¾ to 20 hp single phase motors include open protected, totally enclosed fan cooled and explosion proof frames. A treatment which makes the motors suitable for use in damp, dirty or corrosive atmospheres is covered, as well as tabulated ratings, dimensions and weights. Bulletin 470-B.
Robbins & Myers, Inc., Springfield, Ohio.

Brass Mill Products. This 24-page lexicon of copper and copper alloy terminology provides definitions to such terms as "deoxidized," "flat wire," "patent leveling" and "resquared metal."
American Brass Company, Dept SBR, Waterbury, Conn.

Coatings. Fire retardant coatings for three types of thermal insulation materials are the subject of this four page folder.
Royston Laboratories, Inc., Pittsburgh 38, Pa.

Roof Ventilators. Design and construction features of 13 basic centrifugal type roof ventilator sizes, including 111 different motor and belt-drive combinations are listed for various building environments in Bulletin 4004. A capacity table for the motor-drive combinations includes catalog numbers, quietness rating, motor hp, fan rpm, tip speed, cfm at various static pressures and net weight. Eight pages.
American-Standard, American Blower Div, Detroit 32, Mich.

Transmitter. Bulletin 170 describes in building block style the design features of the MPT motion position transmitter.
Brooks Rotameter Company, Box 432, Lansdale, Pa.

Motor Controls. Catalog 14, illustrated with product pictures, wiring diagrams and dimensional drawings, contains size, weight and rating information on motor controls. 80 pages.
Arrow-Hart & Hegeman Electric Company, Industrial Control Div, 103 Hawthorne St., Hartford 6, Conn.

Mechanical Dehumidifiers. Illustrating the company's line of dehumidifiers using refrigerant compressors, to a dewpoint of 36 F, and discussing means of achieving dewpoints to −25 F, this bulletin covers unit capacities from 640 to 11,856 cfm, from 5 to 60 hp refrigerant compressors. 32,032

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Why lose dollar-making storage and selling space to house compressors, when you can now use the Kramer Outdoor Compressor? The THERMOBANK COMPRESSOR SYSTEM is factory assembled, tested and run-in; it saves installation time and money.

The refrigeration industry again finds Kramer blazing a new path with its THERMOBANK SYSTEM—the original and only automatic re-evaporating, non-overloading, fastest hot gas defrost system—now engineered to perform unfailingly, not only in a heated space but in any unheated space or outside at any outdoor temperature.

WRITE FOR BULLETIN TC 406A

KRAMER TRENTON CO. Trenton 5, N. J.

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to 1,591,968 grains of moisture may be removed per hour.
M. Blazer & Son, 173 Market St., Passaic, N. J.

Heating, Air Conditioning Equipment. Features of box-fin baseboard radiation, industrial and commercial radiation, motorized zone control valves, co-axial refrigerant condensers and finned tubing are detailed in this four-page brochure.
Edwards Engineering Corporation, 101 Alexander Ave., Pompton Plains, N. J.

Electronic Temperature Controllers. Specification S1010-6 describes the Versa-Tran electronic temperature-controllers for industrial heating and cooling operations.
Minneapolis-Honeywell Regulator Company, Wayne & Windrim Aves., Philadelphia 44, Pa.

Protective Coatings. Synthetic rubber compounds for sealing, glazing and caulking are covered in this 12-page indexed bulletin.
David E. Long Corporation, 220 E. 42nd St., New York 17, N. Y.

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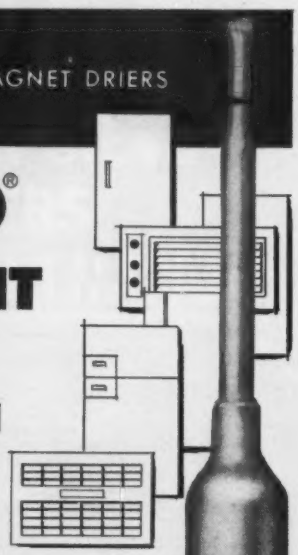
Now KMP®

LITTLE GIANT DRIER with TRIPLE FILTRATION

for General Field Service Work on hermetic and open type units



- **Here is the smallest, most powerful, easiest-to-install drier made.** It will eliminate the worst moisture, screen-out sludge and foreign particles—conditions encountered in general field service work. Though small, the "Little Giant" has ample capacity to handle up to ¾ hp with F-12. "Little Giant" desiccant charge is 100% molecular sieve.
- **Driers are completely dehydrated** and the ends fused and sealed to insure dryness. KMP's undercutting feature permits tubes to be quickly snapped off, as well as cut off, swedged, flared or bent to take care of all field applications from capillary sweat connections to ¼" flare.

Also available

is a complete line of KMP silica gel and molecular sieve moisture magnets. A minimum of models will handle ¼ to 10 ton units. All are backed by KMP—leading supplier to over 80% of the original equipment manufacturers.

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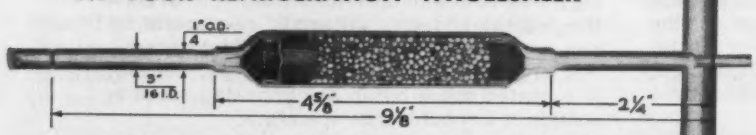


KMP KENMORE MACHINE PRODUCTS, INC.
LYONS, NEW YORK
Driers • Accumulators • Accumulator Driers • Strainers • Capillary Assemblies

Valves, Fittings and Supplies. Pipe nipples, hanger fittings and other new items have been added to this latest edition of a catalog showing valves either singly or complete with screwed or welding-neck flanges. In the back are eight pages of tables on refrigerants, brines, steel pipe, copper and steel tubes, storage temperatures, mechanical equivalents and thermometer scales.
Frick Company, Waynesboro, Pa.

O-Ring Seals. Design and applications of O-rings is the subject of 20-page Bulletin AD-148. Materials of construction, dynamic and static applications, use of back-up or non-extrusion rings and groove-design are included.
Garlock Packing Company, 432 Main St., Palmyra, N. Y.

Phenolic Products. Brochure CDC-358 covers a complete line of phenolic resins, varnishes and molding powders in eight illustrated pages. It includes product features, special properties and detailed technical data.
General Electric Company, Chemical Materials Dept., 1 Plastics Ave., Pittsfield, Mass.

Cooler Door. Cold storage doors designed for use in dairy applications are described and illustrated in this four page brochure covering construction details on sliding dairy doors, standard cooler and freezer doors, vertical lift cooler and freezer doors.
Clark Door Company, Inc., 512 Hunterdon St., Newark 8, N. J.

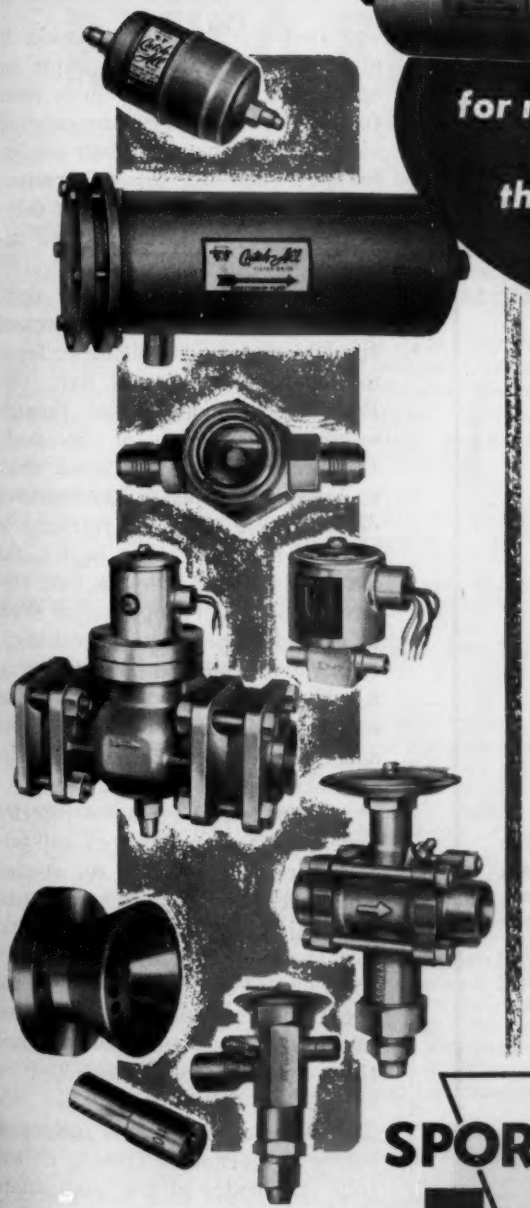
Corrosion Control. Solutions to the scale and corrosion problem are offered in this circular listing several products for correction of specific problems.
D. W. Haering & Company, Inc., P. O. Box 10337, San Antonio 21, Texas.

All-Weather Motors. Vertical hollow shaft motors in ratings of 1 to 30 hp are described in Bulletin 2500.
Louis Allis Company, Milwaukee 1, Wisc.

Remote Control Positioning Switch. Bulletin 357-3 and Technical Data Form P-75 give illustrations, specifications and application descriptions of a positioning switch used primarily for remote control of pneumatic valves or motors, such as those on dampers and in air conditioning units.
Powers Regulator Company, 3434 Oakton St., Skokie, Ill.

Every Air Conditioning Installation *deserves* SPORLAN PEAK PERFORMANCE... *Right Down the Line*

for no matter how large or small the installation... there's a Sporlan combination engineered to fit it PERFECTLY!



Sporlan Catch-All filter-driers with the famous molded porous core, keep systems perfectly clean, perfectly dry and acid free!

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The superior design of **Sporlan Solenoid Valves** with Blue Seal Coils gives high MOPD ratings, permitting them to be used on applications with either refrigerants 12 or 22.

The Flow-Master element, exclusive with Sporlan Thermostatic Expansion Valves for air conditioning, minimizes valve hunting and smooths out evaporator performance.

Sporlan Refrigerant Distributors with the versatile interchangeable nozzles, not only permit a wide range of coil applications, but give uniform distribution to any number of circuits regardless of load or evaporator temperatures.

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Our 25th Anniversary... 1934-1959

EVAPORATOR FROST PERFORMANCE

(*Continued from page* 90)

coil will hold, and yields an overall result of a longer frost block time with a net gain in refrigerating effect.

Table I gives a comparison of results showing for each case the amount of fin surface, time for the return air to reach zero degrees, time for coil to frost block as observed (see Fig. 8) and the return air and product temperatures at that time, the lowest average product and return air temperatures obtained and the average product temperature at zero degree return air point. These values reflect the additional capacity gain of the staged fin coil. The same table shows an improvement ratio of frost block time of 7.5 to 1 as compared to the 4 fin per in. coil in Test 1.

The amount of defrost water collected shows an increase of about 5.7 to 1 over the 4 fin per in. Defrost time in each test was nearly the same in spite of the differences in the amount of water collected. This accounts for the greatly improved defrost cycle efficiency in the 1-2-4 fin per in. coil. The theoretical number of required defrost cycles required per day as shown in Table I is based on defrosting when the return air warms up to zero degrees.

Using values thus obtained the sensible heat gain per day was calculated using the previously obtained defrost cycle efficiency based on the defrost heater input of 732 watt for the defrost times shown. This is further expressed as a percentage of daily condensing unit capacity. These figures are purely empirical and should not be taken to reflect anything but a direct comparison among these tests.

Conclusion—These tests make it apparent that through proper fin space staging, to allow more room for frost collection on the entering air side with progressively smaller fin spaces for the stages thereafter, the result is a coil that first dehydrates then cools the air and improves performance in a gravity type ice cream display case along the following lines: (A) Increased frost block time resulting in fewer number of defrosts per day. (B) Possible savings in cost through more effective use of fin stock. (C) Improved refrigerating effect on top exposed load. (D) Improved defrost cycle efficiency resulting in a smaller defrost heat load to the refrigerating unit. (E) Less possibility of ice cream thermal shock as caused by premature frost block. (F) More likelihood of maintaining a cold air seal over the top opening and thus reduced the effect of frost accumulation on the interior display areas of the cabinet.

Although the results reported here are based on gravity coil performance, the idea of fin staging could well have a place in forced convection low temperature work.

It should be added that the fin spaces reported in this test do not necessarily represent the optimum selections. Until further investigations are reported, it is hoped the results of this investigation will provide guidance for achieving better performance from a fin and tube evaporator at low temperature where humidity is an important factor.

IT'S HERE! "King Zeero's"

NEW REVOLUTIONARY PURE WATER COOLER

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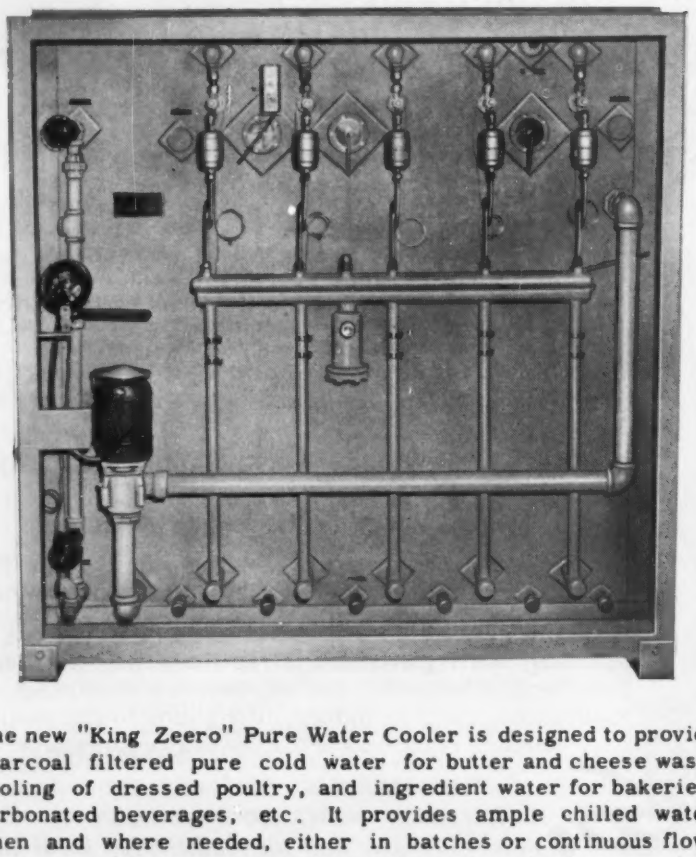
Just What You've Been Waiting For



The new "King Zeero" Pure Water Cooler is designed to provide charcoal filtered pure cold water for butter and cheese wash, cooling of dressed poultry, and ingredient water for bakeries, carbonated beverages, etc. It provides ample chilled water when and where needed, either in batches or continuous flow. Ten years of constant experimentation and testing has gone into this cooler to meet "King Zeero's" rigid standards and the approval of the dairy industry.

Write for details to meet your requirements.

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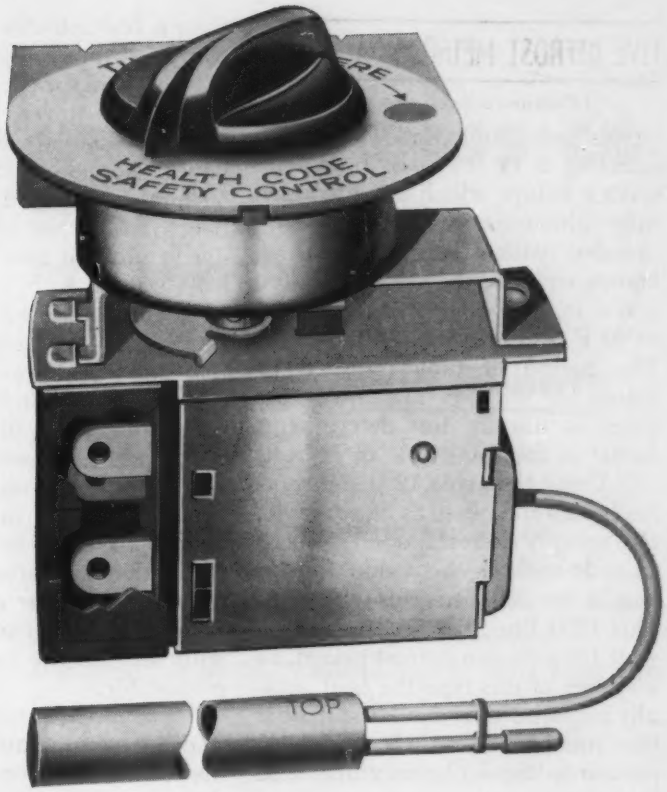
Manufacturers of Ice Builders - Ice Builder Cabinets - Ice Banks

NEW HEALTH CODE SAFETY CONTROL

for coin vending machines dispensing cold or hot items

RANCO F11



Recent changes in U. S. Sanitation Ordinance and Code suggest that vending machines with storage compartments for readily perishable food and beverage items be equipped with controls to insure maintenance of temperature *not higher than 50° F.* (cold items) nor lower than 150° F. (hot items). Only exception is a 30-minute recovery period following loading or servicing, after which time the system must have returned and be in normal temperature operation.

Ranco has developed two controls (F11-1000 for cold vending and F11-1200 for hot vending) which automatically prevent machines from vending food or beverages at unsatisfactory temperatures. They are in addition to, and do not interfere with, the regular cycling thermostat required in the machine.

PRESCRIBED SAFETY VALUES

F11-1000 (cold vending): prevents mechanism from dispensing when refrigerated compartment temperature rises to 50° F. or above.

F11-1200 (hot vending): prevents mechanism from dispensing when heated compartment temperature drops to 150° F. or below.

30-MINUTE RECOVERY PERIOD: Both controls have a 34 (±4)-minute time range, non-adjustable.

OPERATION OF THE CONTROL

The F11 Control is a semi-automatic type with manual reset mechanism responding to temperature change and is equipped with a mechanical, spring-wound timer as a component.

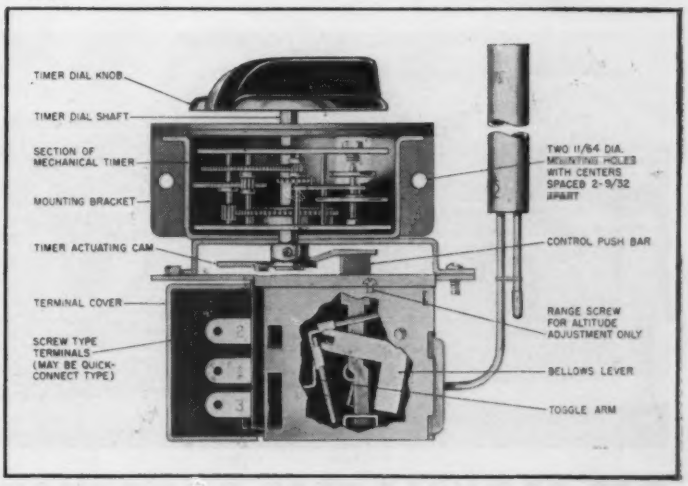
AUTOMATIC FEATURE of the control prevents coin mechanism from dispensing items when temperature of the compartment reaches the prescribed safety setting.

MANUAL FUNCTION is through the mechanical timer (not made by Ranco). Dial knob must be turned to an indicated stop to start the timer after loading or servicing. An actuating cam-push bar mechanism closes the contacts circuited to the machine so it will continue to dispense through the fixed 34-minute recovery period. If the system does **not** recover to normal operating temperature, the push bar unlatches to prevent dispensing. If the system recovers, the push bar remains latched and dispensing continues.

ALTITUDE ADJUSTMENT: correction provided for elevations to 8000 feet.

ELECTRICAL RATINGS: (Listed with UL and Canadian Standards Association) 115 volts a.c. non-inductive, 2000 watts, 7.4 amps. full load, 45 amps. locked rotor. 230 volts a.c. non-inductive, 1000 watts, 3.7 amps. full load, 23 amps. locked rotor.

Special modifications are available to meet your product needs through your Ranco distributor.



See your Ranco Distributor or write direct for complete, detailed Bulletin 1726 with wiring diagrams, technical data on F11 Series controls.

Ranco®
INCORPORATED
COLUMBUS 1, OHIO

More than 100 million Ranco Controls now in use

FIVE DEFROST METHODS

(*Continued from page* 65)

copper, aluminum, and steel and selecting a 12 ft frozen food self-service fixture which has a copper tube aluminum finned coil surrounded with steel plates all of known weights, the value of H_S for a rise in temperature from −40 F to 40 F is approximately 1800 Btu. The period of time required to warm up the coil and wall surfaces is usually the determining factor in the total time of defrost.

Using the same 12 ft self-service frozen food case as an example, if during a defrost period 1 lb of frost is melted the value for Q_d would be 200 Btu plus 4250 Btu plus 1800 Btu = 6250 Btu or 4400 watt for a 25 min defrost period. In a system of this type the heat actually required to remove the frost is the smallest value. It has been proven in the laboratory and field that during any defrost period, regardless of the weight of frost, the actual time required to complete a defrost period does not vary more than a few minutes with the electric heat input being constant at 250 Btu/min for the above example freezer. However, if the coil becomes completely iced so that air flow is stopped, a long time (40 to 50 min) is required before normal heat transfer can take place in order to proceed into the normal defrost time.

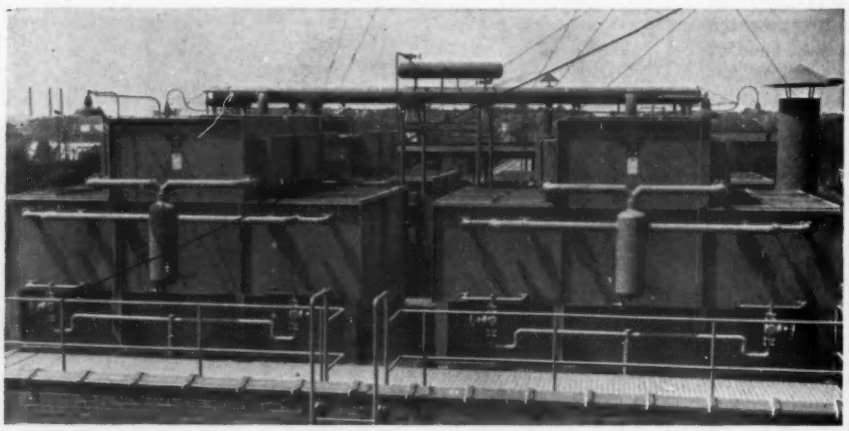
Most walk-in freezer evaporators are of such a compact design that good heat transfer from electric heaters to the frost can be accomplished without the fans running and as a result a more rapid defrost period is obtainable. Since the factor H_A is in direct proportion to the time involved, a more rapid defrost period will greatly reduce this factor and the shorter defrost period can be obtained with less electric heat for a given evaporator.

For medium temperature meat and dairy refrigerators, even though the coils collect more frost, a shorter defrost period is obtainable due to the higher temperature of the metal surfaces at the time defrost is instigated. A period of 15 min two to three times per day is normally required for self-service refrigerators. Since the time period is shorter, the value H_A is greatly decreased and less wattage is required even though more frost is to be melted.

Fig. 7 represents a cross section of a typical modern self-service frozen food case with location of fan, evaporator, drain, electric heater, and air path indicated. Fig. 8 is a temperature vs. time graph of a defrost period for a 12 ft freezer of the type represented in Fig. 7. The thermocouples measured the temperature of the outer surface of one of the evaporator tubes and the water temperature in the drain cup. The drain cup is the last point the water reaches before flowing into the exterior drain pipes.

Controls for any electric defrost system consist of a time clock which must energize the electric heaters, stop the compressor motor, and stop the evaporator fan if required. The termination may be one of three ways: (1) timer terminate; (2) pressure terminate by the rise in suction gas pressure; and (3) temperature terminate by a rise in the evaporator surface temperature or air temperature. The advantages and disadvantages described for controls in the unit-off, natural-defrost method apply to electric defrosting.

Water Defrost employs a spray of water utilizing the sensible heat of the water as a heat source. The drains are usually electrically heated for this system. The water is circulated by a pump controlled by a time clock. The timer stops the compressor during defrost and energizes the electric drain heater. Water has an excellent thermal capacity available for defrosting. A 1 ton evaporator normally requires 7 gpm for a period of not longer than 5 min. In analyzing the thermal capacity of water, it is of interest that if 7 gpm of water flow for 5 min and have a temperature change of only 10 degrees in passing over the frost and melting it, 10 kw of electricity would be required to do the same work – assuming that there was no difference in the loss to the air. Obviously, introducing 10 kw to a 1 ton coil would tend to damage the



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NIAGARA AEROPASS CONDENSERS give you trouble-free, automatic refrigeration at the least spread between head-pressure and suction pressure. You gain a great power saving. You also get removal of superheat before condensing, condensing at subcooled temperature and a refrigerant fully purged of oil. Out-door air takes up heat of condensation through evaporation of the least amount of water; low temperature condensing means freedom from scaling and loss of capacity. Niagara sectional design offers you lower costs for more compact equipment, easier to keep up. You get always full capacity, "new plant" efficiency and continuous savings such as 95% of condensing water cost that add to your profit. Managers who know their costs buy Niagara Aeropass Condensers.

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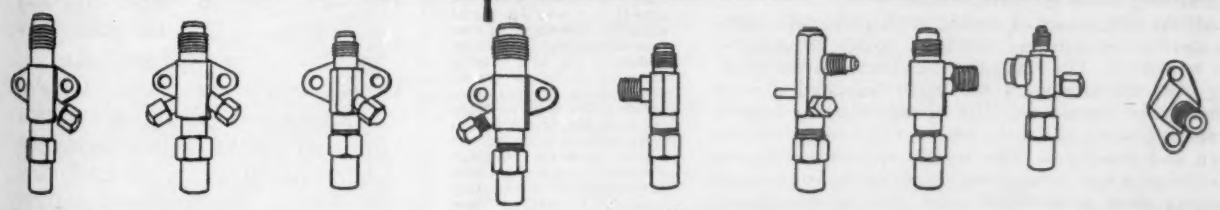
Niagara District Engineers in Principal Cities of U. S. and Canada

MUELLER BRASS CO.

introduces . . . hydrogen brazed steel compressor valves that combine high quality with low price

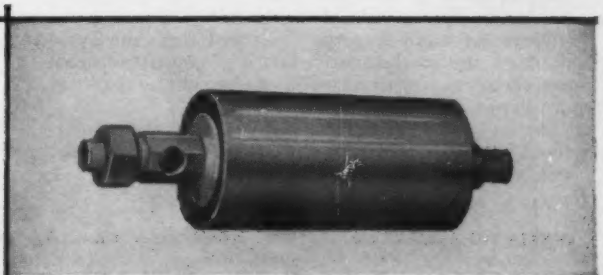
A new line of Streamline® Hydrogen brazed steel compressor valves is now available to you at amazingly economical prices. They incorporate the same high quality construction as the forged brass valves developed and introduced by The Mueller Brass Co. many years ago. The combination of steel construction, and new Hydrogen Brazing techniques has made possible the production of sturdy, dependable valves that will give outstanding performance in your products, yet hold down those all-important costs.

Included in this new line of valves are compressor valves, (single or double port and adjustable), steel evaporator liquid valves and receiver valves. All are available in both flare and solder type connections. Write us for full information on cost and engineering specifications . . . a Mueller Brass Co. Sales Representative will be glad to call on you.



A NEW COMBINATION . . .

Mueller Brass Co.'s new exclusive combination filter-drier and liquid indicator assembly is also a hydrogen-brazed all steel product. It combines the perfectly balanced DRYMASTER Filter-Drier and the easy-to-read SIGHTMASTER liquid indicator into one economical, compact, quickly installed unit.



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MUELLER BRASS CO. PORT HURON 15, MICHIGAN

coil; consequently, it is necessary to introduce electric heat at a lower rate for a longer time.

Water defrost systems are most applicable to large evaporators for walk-in freezers and coolers.

Other External Heat Source Defrost. Additional heat sources are possible by using a fluid such as glycol as a heat transfer vehicle. This secondary fluid is pre-heated by electricity, steam, or other methods to add sufficient quantities of heat to obtain rapid defrosting. Where the heat is applied by circulating the secondary fluid in an inner tube of the evaporator coiling, a rapid defrost is accomplished with a minimum of heat lost to the surrounding air.

This method is most applicable to large low temperature evaporators as used in frozen food walk-in's.

Defrost controls for a system of this type must include a time clock which will stop the compressor, start the fluid (glycol) circulating pump, turn on the heat applied to the fluid, and then terminate defrost similar to other methods discussed for other systems.

SUMMARY

Commercial refrigerators used in modern food markets utilize numerous defrost systems with many variations of controlling the time of defrost. In the design or selection of a method, it is well to consider the following factors:

Type of refrigerator installation giving special consideration to long lengths of multiple cases, walk-in storage refrigerators, and self-contained refrigerators.

Temperature maintained in the refrigerator with due consideration given to the temperature of the drain areas.

Durability of the perishable products displayed or stored when subjected to temperatures above their critical points.

Complexity of system with respect to installation and cost.

Improved efficiency of the complete refrigeration system with fewer and shorter defrosts.

Available heat sources and method of transferring to frost.

Actual heat required to melt frost is relatively small when compared with heat lost to air and heat necessary to warm the coil and wall surfaces above 34 F.

Heat lost to air is in direct proportion to duration of defrost time and becomes a large factor in long defrost periods where air is utilized to transfer heat to the frost.

Time required to warm coil and wall surfaces is determining factor in total defrost period.

Time required to warm coil and wall surfaces being a constant for a given system and being the determining factor in total time of defrost means that regardless of amount of frost, the total time of defrost is fairly constant for a given system unless the coil is iced to a solid point permitting little or no air circulation.

Application of time controls to use pressure or temperature terminating means is a solution to varying icing conditions.

REFERENCES

"Latent Heat Source For Hot-Gas Defrosting" by Paul Reed, *Air Conditioning & Refrigeration News*, May 14, 1956.

"Automatic Defrost" by Otto J. Nussbaum, REFRIGERATING ENGINEERING, July, 1957.

Now check superheat
...EASIER...FASTER...BETTER

Now comes a revolutionary development in superheat testing. The kit illustrated here gives you the easier, faster, and above all the *more accurate* superheat readings you need for that all-important adjusting and setting of the expansion valve.

All the difficulties of testing with glass tube thermometers—positioning, reading, costly breakage—are wiped out. The small bulbs of these distant reading *dial* thermometers are easily attached exactly where they should be. The distant reading feature permits placing the dials where they can be readily seen and compared. The widely spaced *one*-degree markings in the testing zone assure far more accurate reading than is possible with the closely-spaced markings of glass tube thermometers.

Note the many features described opposite—particularly the method of insulating against ambient temperature . . . which assures more accurate readings . . . better results.

At their moderate price you can own these long-lived "Super-heat" Thermometers at a fraction of the cost of using the breakable, short-lived kind!

Write for facts or ***See Your Wholesaler***

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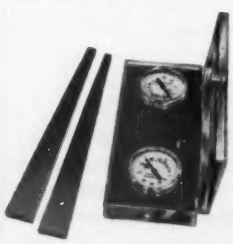
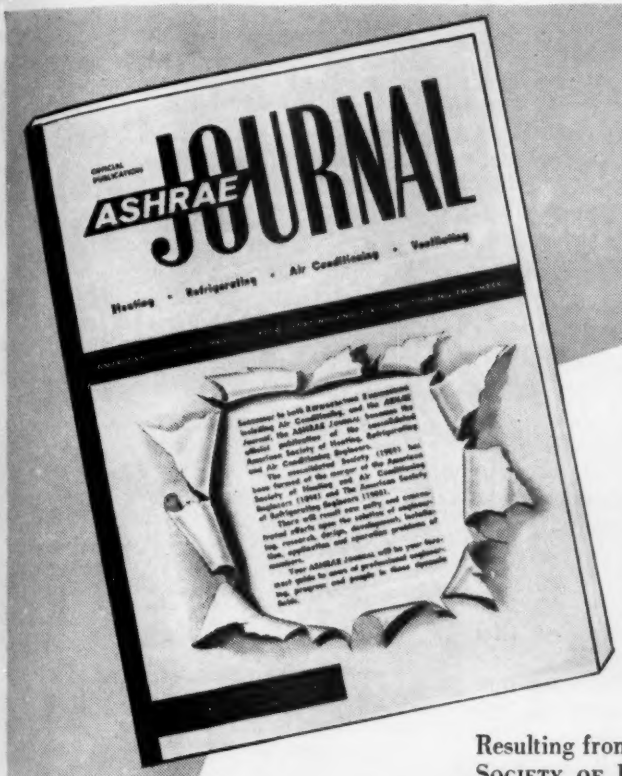
MARSH Refrigeration Instruments

The "Super-heat" testing kit contains two identical, 2-inch dial-size thermometers (one for evaporator inlet and one for outlet) in attractive, highly polished brass cases. Dials are graduated −40° to +65° F. Widely spaced *one*-degree calibrations in testing zone are *guaranteed* accurate within *one* marking plus or minus. (Glass tube thermometers have crowded markings.) Each thermometer has a 12-inch capillary tubing. Bulbs are small for use in tight quarters. Kit contains two non-absorbent insulating wrappers for easy attachment of bulbs to evaporator tubing . . . also to insulate bulbs from ambient temperature. (Field studies have shown that for the highly accurate readings essential to superheat testing, ambient temperature must be insulated from thermometer bulbs.)



Thermometers and "wrappers" are neatly held in durable, transparent plastic box with convenient hinged cover, measures only 3" x 8" x 1¼".



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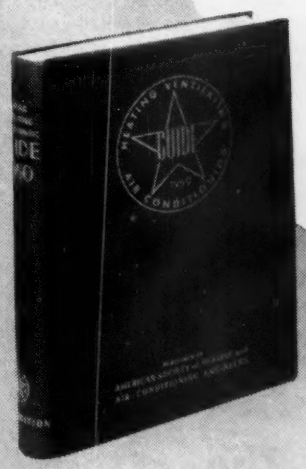

CIRCULATION DATA

ASHAE Members	12,084
Plus the ABC Circulation of "Refrigerating Engineering Including Air-Conditioning"	10,296
Making a TOTAL CIRCULATION of	22,380

For complete details on circulation, rates and mechanical requirements contact:

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FInancial 6-7255

AMERICAN SOCIETY OF HEATING REFRIGERATING AND AIR-CONDITIONING ENGINEERS
62 WORTH STREET, NEW YORK 13, NEW YORK

PRODUCTS

(Continued from page 116)

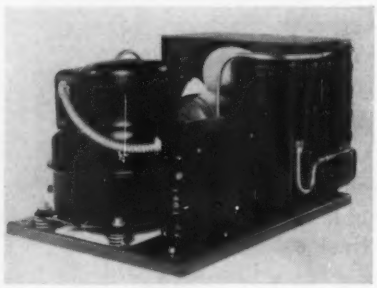
the instrument's built-in electromagnetic pick-up may be attached to an oscilloscope or oscillograph by means of two binding posts on the meter.

In a full frequency range from 120 to 15000 cpm, it is accurate to within 10% at all frequencies with a weighted reed (120 to 600 cpm) and to plus or minus 3% with a free reed (600 to 15000 cpm). It weighs one lb and measures 9¼ x 1⅝ x 1 in.
Korfund Company, Inc., 48-53D 32nd Place, Long Island City 1, N. Y.

SINGLE FAN CONDENSER

Two compact 1 hp, Refrigerant-12 condensing units with single fan construction are in capacities comparable to those obtained with unit of the twin fan construction.

Single fan construction provides a compact unit with limited mounting space needs. The over-all height of



the unit remains the same as the twin fan one; width and length are reduced. Models BRL100T and BRH-100T feature twin cylinder motor compressors designed to operate on 230 volt, single phase or 208/220 volt, three phase current, with Refrigerant-12.
Bendix-Westinghouse Automotive Air Brake Company, 950 E. Virginia St., Evansville 11, Ind.

LOW-HEIGHT FURNACES

Designed for basement or utility room installation, where headroom is at a premium, these Lo-Boy furnaces may be adapted to summer cooling. The series includes six sizes with inputs ranging from 75,000 to 200,000 Btu, and features a Stratosteel heat exchanger, ceramic-coated.
Carrier Corporation, Syracuse 1, N.Y.

DAMPER CONTROL

New damper and quadrant regulators offer methods of controlling dampers to prevent rattles and air leakage. Constructed of heavy gauge cadmium plated metal, all dial dampers and quadrant damper regulators may be mounted on either round or flat surfaces. The complete line ranges from ¼-in. damper regulator sets to ½-in. quadrant damper regulator sets for positive control in small dampers or in heavy duty heating, ventilating, blower and exhaust systems.
Elgen Manufacturing Corporation, Regulator Div., 32-49 Gale Ave., Long Island City 1, N. Y.

EXPANDED LINE

Oil furnaces in this manufacturer's line include three series with several models in each, ranging in capacity from 84,000 to 166,000 Btu. The gas furnace line has improved additions this year; bringing the number of models to 29. In both oil- and gas-fired furnaces, the Regulaire, Stylist and Trimline series feature economies in fuel. The Regulaire series is also utilizing a newly developed blower control.

In central air conditioning self-contained horizontal units in the Tuckaway series include a 2 hp model, two 3 hp models, single or double compressors, and a full 5 hp double-compressor model. Remote units in 2, 3 and 5-ton capacities are with a choice of four evaporator components for adding summer cooling to any type of forced pair system and two complete evaporator blower units.

Space heater lines are in the Coronet series—a circulator or radiant in the gas line and in two sizes in the oil line—and the Classic, a newly designed wall furnace.
Perfection Industries, Div. of Hupp Corporation, 1135 Ivanhoe Rd, Cleveland 10, Ohio.

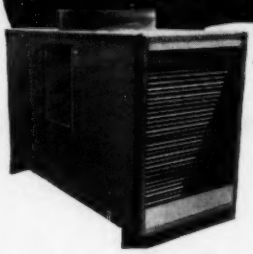
MECHANICAL FILTER

For protection of sensitive gear and equipment, this manufacturer offers the Dustfoe Space Filter, a mechanical type of cleaner which does not utilize an oil treated collection plate, and thereby reduces the danger that vaporization will result in oil being carried through the cleaned air. In 1000 cfm capacity, it is 24 x 24 x 6 in.
Mine Safety Appliances Company, 201 N. Braddock Ave., Pittsburgh, Pa.

PACKAGED CONTROL

Complete units with temperature and contaminant control systems, lighting, electricity and plumbing are pre-engineered for use where rigidly controlled environment is necessary for testing and assembly. Temperatures may be maintained at plus or minus ½ deg; humidity at plus or minus 2% relative. Filters remove airborne con-

Do Your Specs Call for "WOOD COOLING TOWERS?"

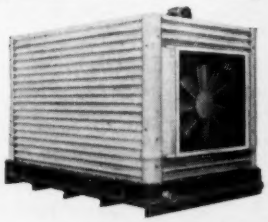


Marley Makes em All

Marley Permatowers®

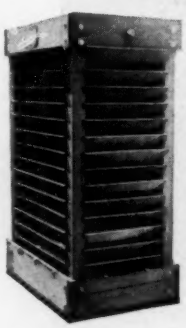
In addition to Marley's complete line of steel packaged cooling towers (see facing page), Marley makes a complete line of packaged wood towers for those jobs where unusually corrosive conditions are of paramount importance, and other models are the last word in first-cost economy.

Newest of the wood line is the PERMATOWER that sets a new standard of cooling tower



Marley Wood Aquatowers®

durability through the liberal use of inert materials throughout. Double-Flow design principle makes them adaptable to restricted locations. WOOD FORCED-DRAFT AQUATOWERS, with durable redwood structures and asbestos cement board casings, are also long-lived towers and top discharge solves many location problems. SPRATOWERS combine engineered, plus-performance spray system and design simplicity for best natural draft cooling and rock bottom economy. Your Marley distributor can supply these products or complete application information.



Marley SpraTowers®

The Marley Company

MARLEY

Do Your Specs Call for "STEEL COOLING TOWERS?"

Marley Aquatowers®

Marley Makes 'em All

(hot dip galvanized after fabrication)

Marley AquaCoolers®

MARLEY AQUATOWERS®

are the world's original and most popular steel packaged water cooling towers that offer user and contractor everything that spells *complete satisfaction*—performance, economy, minimum maintenance and lasting durability.

Fourteen models with capacities from three tons nominal and up permit selection of an Aquatower ideally suited for any specification. For larger capacity jobs, Marley offers twin Aquatowers served by a common cold water basin, that feature operational flexibility, permit "sensitive" cold water temperature control.

So, just name the time, the place, the gpm, and special installation requirements — your nearby Marley distributor will furnish a steel Aquatower that fits the job as though it had been custom-designed. All models feature Marley-designed mechanical equipment, lifetime wood filling, ease of disassembly (if necessary). Casings and multi-blade fans are hot dip galvanized after fabrication.

MARLEY AQUACOOLERS®

feature quiet operation, full performance and installation in minimum plan area, making them top choice for indoor service—and equally adaptable to location out of doors. AquaCoolers are simply and economically vented and their centrifugal blower fans provide operating pressures that overcome draft loss developed by ductwork.

Balanced spray system uses Marley non-clog nozzles that minimize maintenance. Engineered spray water break-up and ample depth of MarPak filling deliver air-water contact that means full performance in every capacity size.

Ten models, nominally from 5 tons and up, are available from your Marley distributor. Smaller models are shipped completely assembled; larger ones reach your job in two sub-assemblies that are readily bolted together. Heavy, braced casings, basins and fan scrolls are hot dip galvanized after fabrication.

The Marley Company

Kansas City, Missouri

taminants to five micron size.
Moore & Hanks Company, 9702 East Rush St., El Monte, Calif.

CONDENSING UNITS

Commercial and industrial refrigeration condensing units, ranging in size from 7½ to 150 ton capacity, may be the shortest units on the market today. (The smallest unit is 54 in. long.) They are designed for industrial plants, supermarkets, apartment buildings, hotels and lockers.
Bell & Gossett Company, Morton Grove, Ill.

PACKAGED HEATERS

Five oil and three gas winter air conditioners designed with warm air blowers positioned under the heat exchanger are offered in this 1959 line. The new blower arrangement makes possible a cabinet of minimum height without interfering with heating efficiency.

Btu/hr input ranges from 100,000 for a gas-fired hi- or lo-boy and 224,000 units for a suspended oil model.
Thatcher Furnace Company, Garwood, N. J.

SEALING MATERIAL

To seal parts exposed to high humidities, high-frequency freeze-thaw cycling and those subject to complete submersion in water, this manufacturer offers a sealer for the effective prevention of air or moisture passage between similar or dissimilar materials. Identified as Wat-R-Bar, the sealing material will not flow from −40 to 400 F and will not become brittle at −40 F.
Presstite Engineering Products Company, 39th and Chouteau, St. Louis, Mo.

You can't wrap the system in cellophane...

PROTECT IT WITH BANOX®

Forms a molecular film on all metal surfaces contacted by recirculating water. Condenser, pump and water lines are protected against corrosion. Use at Spring start-up — after acid cleaning — at shut down. Get Banox from your Refrigeration Wholesaler today.

CALGON COMPANY
DIVISION OF **HAGAN** CHEMICALS & CONTROLS, INC.
HAGAN BUILDING, PITTSBURGH 30, PA.
In Canada: Hagan Corporation (Canada) Limited, Toronto

PLASTIC SEALANTS

Sealants with high bonding strength, based on Thiokol polysulfide rubber, are for use on metals, glass, stone, concrete and wood. They are 2-component systems which mix readily and are applied in a pasty non-flowing consistency. When cured, sealants will set to a permanently firm flexible, non-tacky synthetic rubber.
H. B. Fuller Company, 255 Eagle St., St. Paul, Minn.

CONTROL VALVES

Split-body valves for use in high-temperature, high-pressure control applications are in large flow capacities and face-to-face dimensions based on ISA standards. They may be interchanged with any standard make of diaphragm valves. 432 different valve combinations may be developed from a single split-body.
Kieley & Mueller, Inc., 64 Genung St., Middletown, N. Y.

CAPILLARY CONTROL VALVE

For use in reverse cycle heat pumps, this valve is installed in conjunction with the capillaries between the high and low side of the system. It provides single capillary restriction for the cooling cycle and a double capillary restriction for the heating cycle.

The valve body is made of hot forged brass and hard drawn copper tubing, silver brazed. The action employs a Swedish spring steel disc and an alnico permanent magnet. A built-in 80-mesh monel strainer affords maximum protection to the capillary tube.

The capillary control valve is applicable to all units up to and including five ton and may be used for either single or multiple capillary installations.
Watsco, Inc., 1020 E. 15th St., Hialeah, Fla.

hot-dip GALVANIZING

Quality work to rigid standards, emphasizing quantity production on "hard-to-do" jobs. Also experienced metal-spraying applicators. Refrigeration, air-conditioning and heating items our specialty. Ideally located for eastern seaboard and export shipments.

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Add years of life to metal by sealing in MOLTEN ZINC

The Southern Galvanizing Co.
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*FIRST LINE *Extended-Surface* HEAT EXCHANGERS

ASK THE AEROFIN MAN

Specify Aerofin and you specify high efficiency, long service life and low maintenance and service costs.

Take advantage of Aerofin's unequalled experience, production facilities, and materials-testing and design research—of Aerofin's complete engineering service at the plant and in the field.

Ask the Aerofin man.

* Aerofin makes extended heat surface exclusively — not as a by-product, not as a side-line. Sold only by manufacturers of fan-system apparatus. List on request.

AEROFIN CORPORATION

101 Greenway Ave. Syracuse 3, N. Y.

ASHAE 65TH ANNUAL MEETING

(*Continued from page 75*)

temperature water heating. Professor W. L. Rogers described the sources of noises in piping systems and how they may be reduced.

Methods for selecting standard pumping equipment for conventional hot water heat and closed water heat systems were offered by G. F. Carlson. Factors which should be considered in the design of refrigerant piping were presented by J. R. Chamberlain. He suggested the use of close coupled factory equipment rather than fabricated piping in the field.

Four speakers covered various factors relating to heat pump performance. F. L. McFadden, Jr., described a water-sink type of heat pump which utilizes water drawn from an adjacent lake. He provided data on actual operation of the installation, in use for seven years in the Southwest, and also related operating problems which have occurred.

J. D. Kroeker reported upon a heat pump installation in the Northwest part of this country. He included data on practice and performance. Package heat pumps, including a summary of the present design of this equipment, were discussed by W. A. Spofford. He also presented operating results obtained from 78 residential installations in various locations. E. R. Ambrose reported operating results obtained from central heat pump installations in various locations. Performance of an air source compound heat pump was described by A. D. Spillman.

Final engineering session of the Meeting was the Symposium on Corrosion and Water Treatment. F. S. Hodgdon outlined sources of corrosion and offered suggestions for overcoming this problem. Both chemical and mechanical methods were discussed. Additives would control or change the acidity contained in the water. Electrolytic corrosion could be minimized by using similar metals and corrosion due to mechanical means prevented by avoiding water traps.

Selection and methods for handling insulation applied to outer surfaces of piping which could reduce this type of corrosion were presented by C. E. Ernst.

L. D. Polderman discussed maintenance against freezing in underground hot water piping. Selection of anti-freeze solutions, keeping solutions at a proper level and some methods to minimize corrosion by these solutions were covered. Preparing specifications properly to provide for the desired solving of water corrosion problems was discussed

GET ACCURATE FLOODED SYSTEMS CONTROL

with this level regulator

A SIMPLE CONTROL IS THE BEST CONTROL

PHILLIPS
FLOAT CONTROLS

- **Proportioning action for smooth feed at all capacities**
- **Tight Closing with Teflon seat discs**
- **Self actuation—no electrical or pneumatic connections needed**
- **Visible liquid level through exclusive "Level Eyes"**
- **Adjustable level achieves maximum capacity with minimum charge**

IN ADDITION — Phillips pilot operated valves are available for all common refrigerants, down to —50° F. Operates with as low as 2 PSI pressure drop and up to 250 PSI with selected springs. Line sizes ½ inch to 4 inches with steel or copper connections.

Solve your flooded system design and application jobs by consulting Phillips. Our firm of engineers have specialized in level control, liquid-vapor separation, liquid circulation and return systems for over 28 years.

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Refrigeration Control Systems
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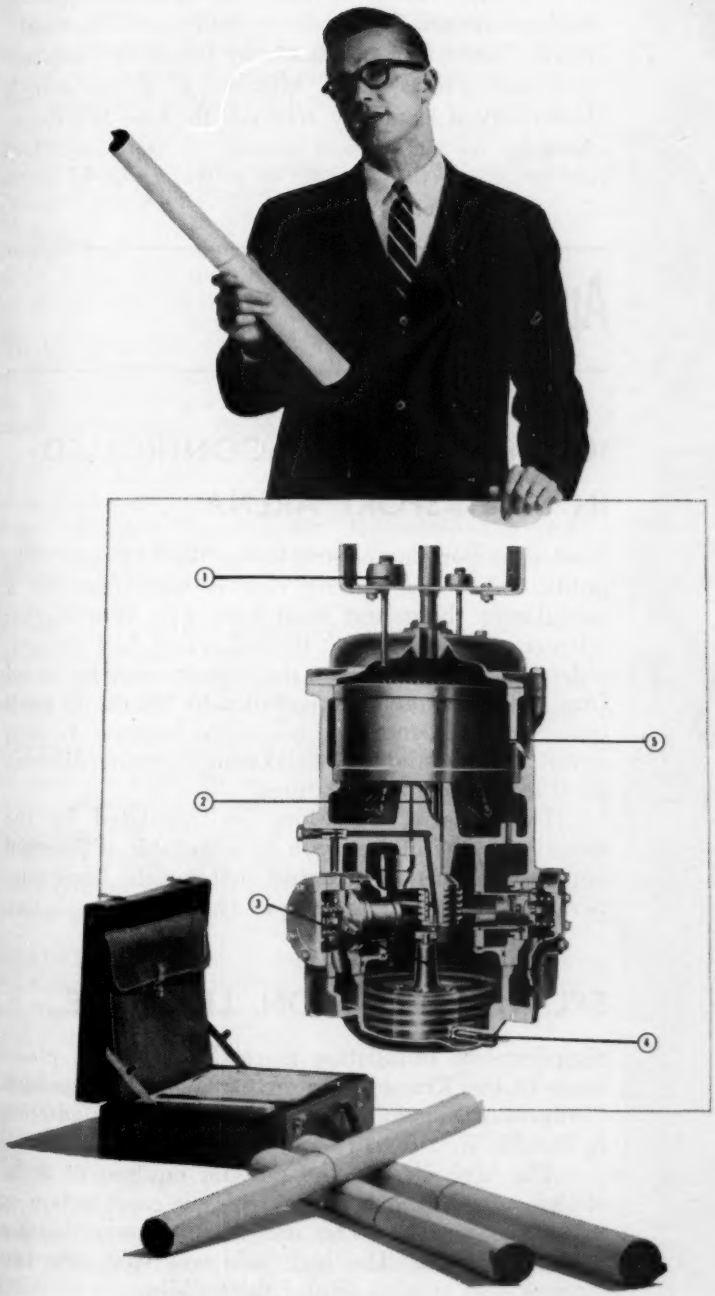


Here are 5 of the 18 reasons why it pays to pick Chrysler Liquid Chillers

Cooling for comfort? Or cooling a process? Only Chrysler Airtemp SW Liquid Chillers offer so many exclusive engineering advantages. Take the hermetically-sealed radial compressor, for example. You get:

① *Floating power.* Rubber-in-sheer suspension absorbs vibration. Noise is gone . . . and so are vibration-caused service headaches.

② *Direct drive.* No belts, no pulleys, no seals to break down. You get greater efficiency, less noise from a more compact package.

③ *Positive lubrication.* Exclusive pressure pump is sealed in a bath of oil. Moving parts get constant lubrication for long, trouble-free life.

④ *Crankcase heater.* Prevents refrigerant from condensing and diluting lubricant. Oil stays clean . . . and always ready to flow.

⑤ *Sealed-in motor.* Dust-proof for longer service life. Directly connected to dynamically balanced crankshaft to cut weight and operating costs.

The rest of the 18 reasons? They include innerfin direct expansion chiller, automatic control panel, easier installation and . . . But why not get the complete story from your local Chrysler Applied Machinery and Systems sales office. Or write: Airtemp Division, Chrysler Corporation, Dept. Z-39, Dayton 1, Ohio.

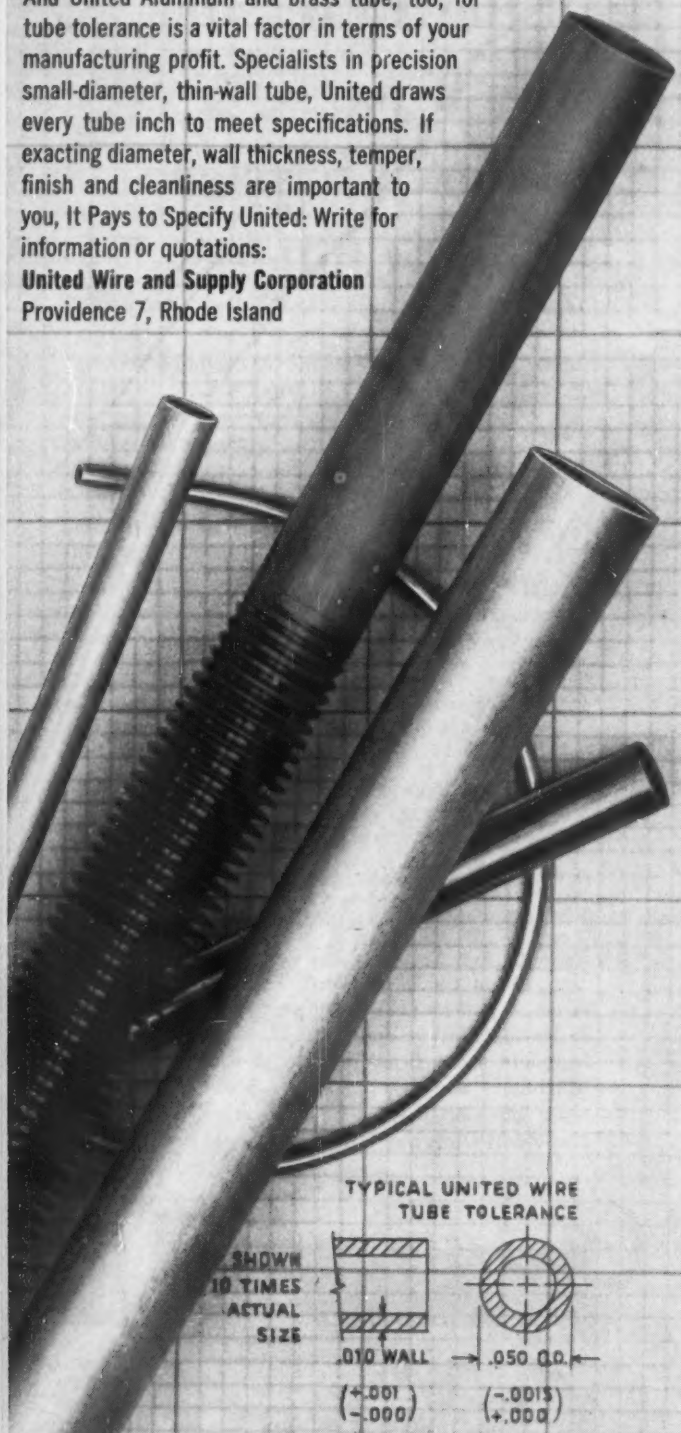
CHRYSLER AIRTEMP

FIRST WITH THE FINEST IN AIR CONDITIONING

TOLERANCE

Reason why — it Pays to Specify
United **Seamless** Copper Tube

And United Aluminum and brass tube, too, for tube tolerance is a vital factor in terms of your manufacturing profit. Specialists in precision small-diameter, thin-wall tube, United draws every tube inch to meet specifications. If exacting diameter, wall thickness, temper, finish and cleanliness are important to you, It Pays to Specify United: Write for information or quotations:
United Wire and Supply Corporation
Providence 7, Rhode Island



For aluminum, brass and copper wire and tube
For silver brazing alloys . . . always **Specify**

UNITED

by W. L. Andrews. Mrs. Olivia Burkey outlined principles to be followed for the proper treatment of water used in air conditioning. Water becomes contaminated from corrosive elements in the air, she said, and suggested the use of chemical additives, testing and keeping equipment clean.

Dr. James Creese, president, Drexel Institute of Technology, addressed the Welcome Luncheon on "Independence Hall and the Kremlin." James Y. Payton was presented the ASHAE-Homer Addams Award for graduate study and Dr. Merl Baker, Director of the Kentucky Research Foundation and Professor of Mechanical Engineering, University of Kentucky, received the Life Members Award.

Applications

ICE TEMPERATURES CONTROLLED IN MULTI-SPORT ARENA

Precise temperature control in the West Orange, N.J., public indoor ice skating rink is maintained by a modulating thermostat used with two Worthington refrigerant compressors. By unloading half the cylinders of each compressor, the capacity may be varied from a full 140 ton of refrigeration to 105, 70, 35 and 0 ton. Varying control is necessary because hockey, figure skating and public skating require different ideal ice surface temperatures.

The capacity variations are regulated by the modulating thermostat with an adjustable differential and motor driven step control enabling the brine temperature to be held as close as ½ deg.

SPLIT INSTALLATION, LESS TIME

Simplicity of installation marked the recent placement of the Kramer-Trenton Company Thermobank Compressor System in the 6000 sq ft freezer addition to Rotelle, Inc., frozen food plant in Pennsylvania.

The high side of the outdoor equipment is installed outside the building with no construction or protection needed. The unit has its own weather protecting hood. The high side was split into two systems, one at each side of the building.

The temperature of the freezer room is held constant with two air-cooled Thermobank compressor systems, each having a capacity of 83,500 Btu/hr. In the three large vegetable and fruit storage rooms are Coolmaster unit coolers.

Less than 100 man-hr were required to install all the refrigeration equipment in the freezer, a 100 x 60 x 12 ft room, capable of storing 30 carloads of frozen food.

WHAT ASHRAE CHAPTERS ARE DOING

(Continued from page 99)

all water systems, or refrigerant systems with self-contained window units. He then compared and contrasted these, using charts to illustrate his lecture.

JOHNSTOWN (H) . . . Reviewing present day developments in energy utilization, H. E. Voegeli, development engineer, American Brass Company, spoke to these members on Solar Energy and the Heat Pump.

After giving a description of early work on heat pumps and energy storage, he described a system he is building in Connecticut using a simplified solar collector, a heat pump and water tank heat sinks.

CINCINNATI (H) . . . Donald Owen talked on Applications of Electric Heating in Our Industry at a recent meeting. He is with E. L. Wiegand Company.

HOUSTON (R) . . . Speaker of the evening in January was H. Banks Edwards, Acme Industries, who covered Heat Pumps and Their Application.

NEW MEXICO (H) . . . Using the ASHAE Guide as a text, C. N. Rink, manager, Industrial Acoustic Corporation, Inc., was the guest speaker here recently. He explained the fundamentals of ventilating acoustics.

PUGET SOUND (H) . . . Recent guest speaker was D. P. Dakos, Pacific regional supervisor, Sound Control Department, Koppers Company, Inc. He presented Predicting the Degree of Noise Reduction Required for an Air Distribution System.

ERIE (R) . . . American Education vs. Russian Education came under scrutiny at the January meeting of this group when Fr. Wilfred Nash, president of Gannon College, presided as speaker.

BUFFALO (R) . . . Air Conditioning and the Power Company was the topic up for discussion at the February meeting, after members toured the Huntley power generating station in the Niagara-Mohawk area.

At the January meeting, the group heard Dennis Lane, Dearborn Chemical Company, on Water Treatment Problems and Solutions.

PHILADELPHIA (H) . . . John Engalitcheff, Jr., delivered an informative talk on Water Conversion Equipment, Theory and Application. He is president, Baltimore Aircoil Company, Inc.

CENTRAL MICHIGAN (R) . . . Scheduled for the February meeting of this group was a discussion of Refrigerant Blown Urethanes, by John J. Rotto, Pellon Corporation.

EMPIRE STATE CAPITAL (H) . . . Miss B. A. Hicks, president, Newark Controls Company, presented a talk on the effects of shock, vibration, acceleration and temperature on the design of aircraft controls.

WESTERN MICHIGAN (H) . . . Mechanics and Operation of the new Monroe Nuclear Power Plant were examined by E. G. Sliper, product engineer, Commonwealth Associates, in meeting with these members.

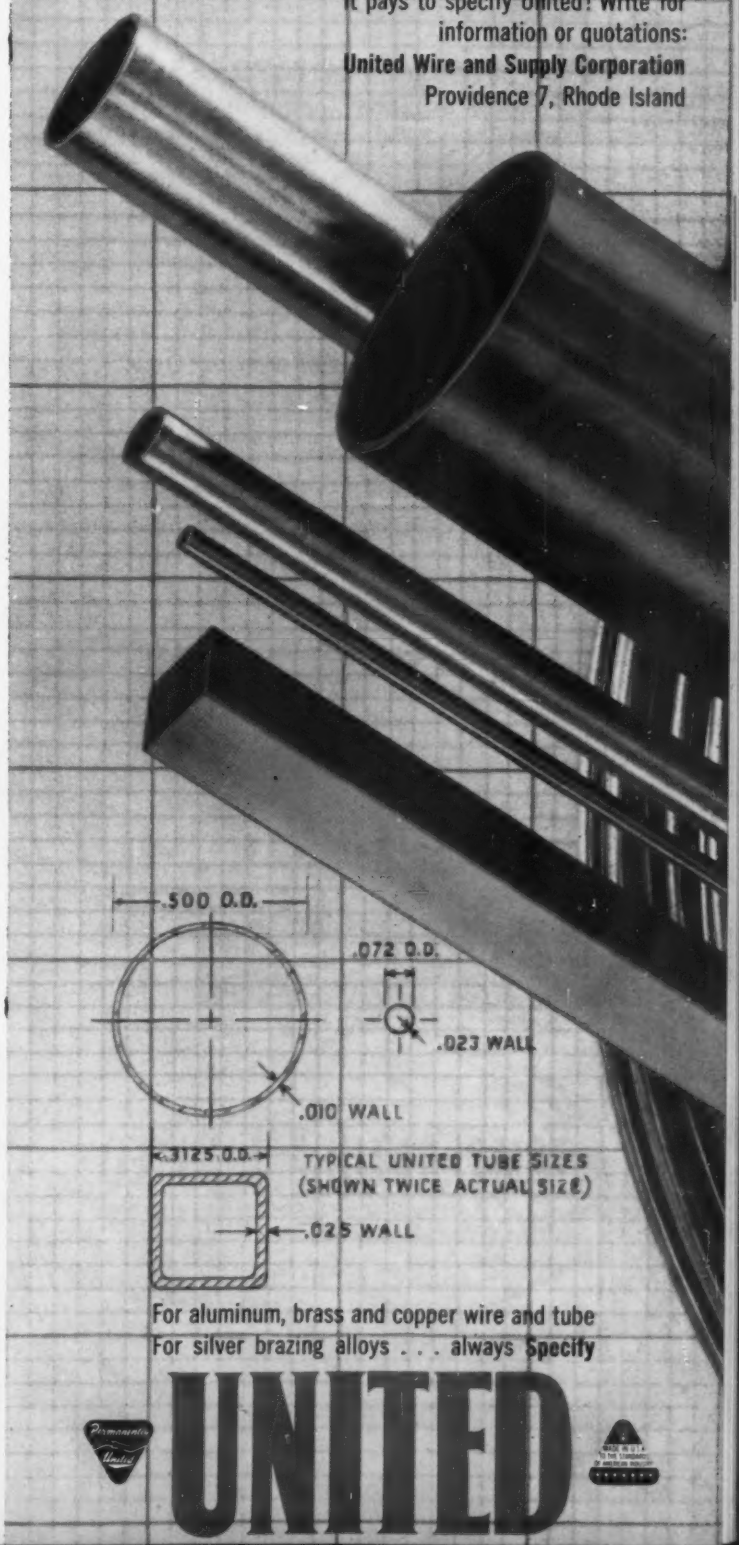
He gave a brief history of the atomic energy developments concerning power plants, indicating he thought it was necessary for several large companies to join forces in order to obtain enough capital for such studies and installations.

SIZE

Reason why — it Pays to Specify United **Seamless** Copper Tube

No matter how "exact" your size requirement for copper, aluminum or brass tube . . . specify United. A precision small-diameter, thin-wall tube mill (from finest capillary up to 1½") United delivers tube to exact specifications for size, tolerance, temper, finish and cleanliness. In coils, standard mill lengths, or cut to your specifications, it pays to specify United! Write for information or quotations:

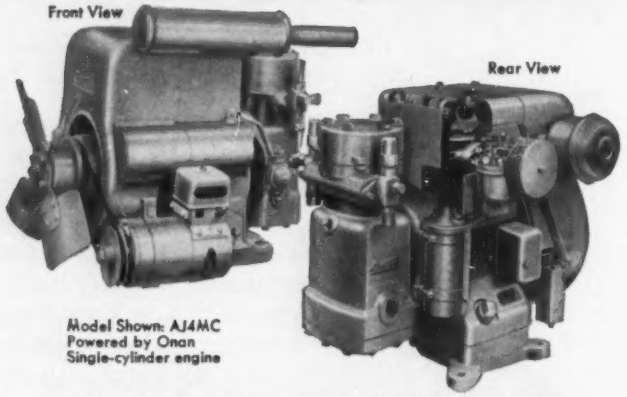
United Wire and Supply Corporation
Providence 7, Rhode Island



For aluminum, brass and copper wire and tube
For silver brazing alloys . . . always Specify

UNITED

AIR CONDITIONING NEWS



Model Shown: AJ4MC
Powered by Onan
Single-cylinder engine

NEW ENGINE-COMPRESSORS

for mobile air conditioning or refrigeration

Built as integrated in-line units with Onan engines direct-connected to Onan compressors. Compact, permanently-aligned and smooth-running. No troublesome belts, couplings or sheaves. Optional accessories: batteries, starters, generators, and fans. Onan 4-cycle engines, built for continuous duty and long life, operate on either gasoline or Propane. World-wide parts and service organization.

THREE MODELS: **AJ4MC:** 1 ton cap., 4.1 H.P., F-12 refrigerant.
LK5MC: 2½ tons cap., 6.25 H.P., F-22 refrigerant.
CCK11MC: 5 tons cap., 12.9 H.P., F-22 refrigerant.

Write for engineering data.

D. W. ONAN & SONS INC.

3766A University Ave. S. E., Minneapolis 14, Minnesota

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LARGEST PRODUCER OF AIR CONDITIONING UNITS OUTSIDE THE U.S.A.

Applicants for and elections to

ASHRAE MEMBERSHIP

In this issue of the ASHRAE Journal there appear two groups of references to current ASHRAE membership.

Because ASHAE has been following the plan of publishing the names of applicants, while ASRE published the names of those actually elected, a dual character, without duplication, here exists.

Subsequent listings, in accordance with ASHRAE By-laws will cover applicants only.

Additions and Changes— ASRE Membership

The following applicants for ASRE membership, promotion or reinstatement were approved by the ASRE Council prior to the consummation of the merger with ASHAE. Symbols indicating the approved grade of membership are *Associate; †Affiliate; ‡Student; all others Member.

PROMOTIONS

Askin, David, Chief Environmental Lab, Frankford Arsenal, Bridge & Tacony Sts, Philadelphia 37, Pa.
Cowan, Armand, Proj Engr, McDonald A C Inc, 261 NW 24 St, Miami, Fla.
†Duerr, Ronald C., Engr, A Burgart Inc, 2705 W Henrietta Rd, Rochester, N. Y.
Jackson, George R., Sls Engr, York Corp, 1221 Concord Ave., Birmingham 13, Ala.
Kalish, Milton, Sls Engr, Dunham-Bush Inc, 250 E 43rd St, New York 17, N. Y.
Kapur, J. C., Mgg Dir, A C Corp Pt Ltd, E 2 Gillanders House, Calcutta 1, India

NEW MEMBERS

(New Basis)

Region VII—St. Louis Section

*Cummins, Robert J., Sls Engr, Marlo Coil Co, 7100 S Grand Ave, St Louis 11, Mo.
†Pryor, Harry H., Proj Engr, National Vendors Inc, 5055 Natural Bridge, St Louis 15, Mo.
†Wadell, T. J., Owner, T J Wadell Co, 1110 S Brentwood, St Louis 17, Mo.

Region IX—Los Angeles Section

†Avila, Victor G., Mech Des, Ralph E Phillips Inc, 2001 W Olympic Blvd, Los Angeles 6, Calif.

Region IX—Pacific Northwest Section

†Raabe, Burlyn C., Engr, Puget Sound Engg Inc, 3838 10th NE, Seattle 5, Wash.
†Walling, V. C., Sls Repr, Virginia Smelting Co, P O Box 337, Bellevue, Wash.

Region IX—San Joaquin Section

Grotzke, William L., Mech Engr, Walter Wagoner & Partner, 1339 Van Ness Ave., Fresno, Calif.

(Old Basis)

Region I—New York Section

‡Blacksburg, Alvin B., Student, City College of New York, 836 Crown St, Brooklyn 13, N. Y.

Region II—National Capital Section

†Caudill, Charles G., Chief Maint Br, U S Army Chem Warfare Labs, Bldg 330, Army Chemical Center, Md.

Region II—Philadelphia Section

*Ikeda, George K., Proj Engr, Borg Warner Corp, York Div, 3 Penn Center Plaza, Philadelphia, Pa.
*Knerr, Charles G., Br Mgr, John J Nesbitt Inc, 8701 Torresdale, Philadelphia 36, Pa.
†Tiffany, Harold E. Jr, Special Repr, Sun Oil Co, 1608 Walnut St, Philadelphia 2, Pa.
†Wood, David H., V P Henry Bower Chem Mfg Co, 2815 Grays Ferry Ave, Philadelphia 46, Pa.

Region IV—Central New York Section

†Soler, Peter J., Draftsman, Carrier Corp, Carrier Parkway, Syracuse 1, N. Y.

Region IV—Montreal Section

†Vince, George J., Serv Mgr, Warren Canadian, 930 Jean Talon W, Montreal, Que, Canada

Region IV—Ontario Section

*Colwell, Denis C., Sls Rep, Clarke-Parry A C Ltd, 322 Howland Ave, Toronto, Ont, Canada
†Hansen, Harry, Sls, Circle Refrigeration Ltd, 50 Soudan Ave, Toronto 9, Ont, Canada

Region IV—Rochester Section

*Fedder, Robert P., Owner, R P Fedder Co, 70 Shorewood Dr, Rochester 17, N. Y.

Region VII—Western Michigan Section

†Heller, Merlgene, Test Engr, Norge Sls Corp, 88 W Broadway, Muskegon Heights, Mich.
Phillips, B. A., Mgr of Research, Whirlpool Corp, 300 Broad St, St Joseph, Mich.
†Travis, Charles W., Partner, Travis-Applegate Co, 714 Building & Loan, Grand Rapids, Mich.

Region VIII—Cincinnati Section

*Worland, Edmond H., Engr Asst, Cincinnati A C Co, 1415 Walnut St, Cincinnati, Ohio

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*Platts, John H., Gen Mgr, Whirlpool Corp, U S Highway 41 N, Evansville, Ind.

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Day, William M., V P, Bryant Mfg Co, 2020 Montcalm St, Indianapolis 7, Ind.

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*Quint, Harold L., Jr Engr, McQuay Inc, 1600 Broadway, Minneapolis, Minn.

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Keith, Fred E., Engr, James A Evans, Cons Engr, 340 Brown-Marx Bldg, Birmingham, Ala.

Region XI—Florida West Coast Section

Jobes, Harry W., V P & Gen Mgr, Typhoon Heat Pump Co, Div of Hupp Corp, 2001 Garcia Ave, Tampa 1, Fla.

Region XI—South Florida Section

Fink, Herbert D., C E, Biscayne Cooling Corp, 610 NE 13th St, Miami 32, Fla.

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†Billhorn, Harry, Instal Forman, Hojalateria Panama, Avenida Transismien, Panama, R of Panama
*Cullen, James C., Mech Engr, Panama Canal Co, Balboa Heights, Canal Zone
*Speier, Claus D., Mgr A C Dept, Elga S A, Via Espana, Panama, R of Panama

Unassigned to Sections

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*Boughner, Homer K., Apl Engr, Kysor Heater Co, 1100 W Wright St, Cadillac, Mich.
†Goltz, George L., Engr, Goettl Bros Metal Prod, 2005 E Indian School Rd, Phoenix, Arizona
*Torres, Jorge, Br Mgr, S A V E R Guinand, 103-63, Valencia-Carabobo, Venezuela

REINSTATEMENT

Lint, Paul R., Mgr, Airtherm Supply Co, 15625 Saranac Rd, Cleveland 10, Ohio
Moe, Gabriel F. Jr, Engr, Philco Corp, Tioga & C St, Philadelphia 34, Pa.
Shine, Thomas P., V P, Joines Rfn Inc, 916 EW Highway, Takoma Pk, Md.

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These are recent candidates for membership or advancement in membership grade as received by ASHAE prior to the consummation of merger with ASRE. Members are requested to assume their full share of responsibility of receiving these candidates into membership by advising the Executive Secretary before March 30, 1959 of any whose eligibility for membership is questioned. Unless such objection is made these candidates will be voted upon by the ASHRAE Board of Directors.

Note: **Advancement* †*Reinstatement*

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Ash, R. S., Pres. & Owner, Temp-Rite Engineering Co., Inc., Tucson.
Bausano, Charles, Sales Mgr., Southern Arizona York Refrigeration Co., Tucson.
Brainard, E. H., Jr., Engr., Refrigeration & Air Conditioning Sales, Inc., Tucson.
Burner, C. A., Jr., Vice-Pres., Clark & Co., Tucson.
Clow, M. J., Vice-Pres., Air Filter Service & Mfg. Co., Tucson

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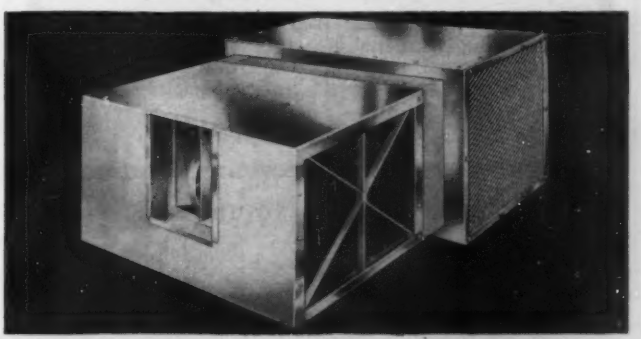

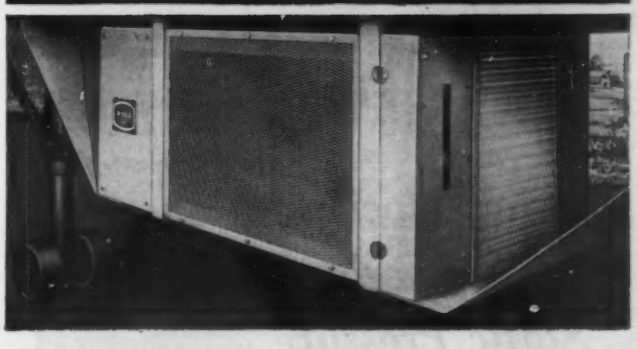

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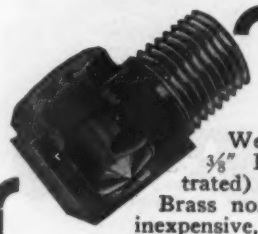

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de Pagter, P. C., Clark & Co., Tucson.
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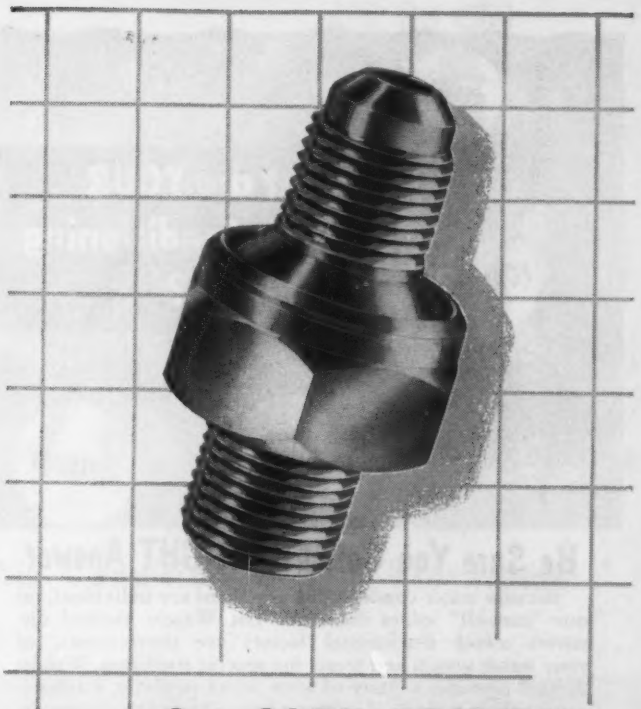

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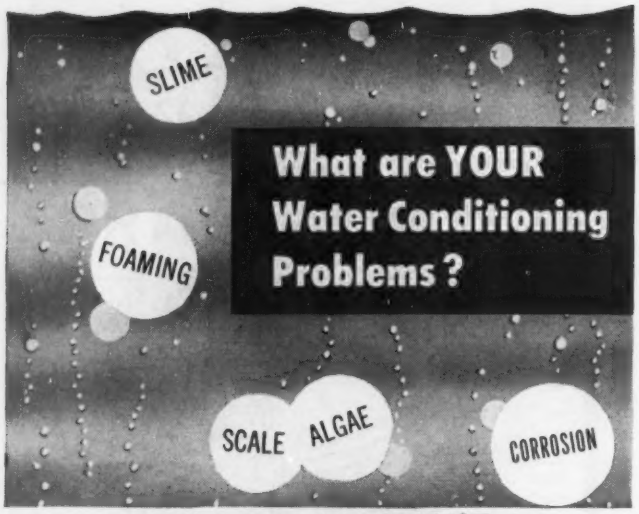

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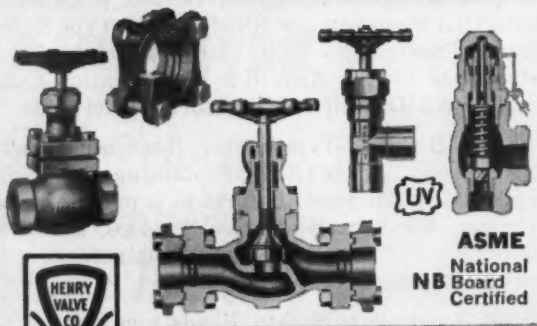

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